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HEAT-POWER ENGINEERING

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BY

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PART I. Thermodynamics and Prime Movers. Third Edition of First Part of Hirshfeld and Barnard's "Elements of Heat-Power Engineering," completely rewritten. 493 pages. 6 by 9. 251 figs. 7 plates.

PART II. Steam-generating Apparatus and Prime Movers, Fuels, Combustion, and Heat Transmission. Third Edition of Second Part of Hirshfeld and Barnard's "Elements of Heat-Power Engineering," completely rewritten. 871 pages. 6 by 9. 828 figs.

PART III. Auxiliary Equipment, Plant Ensemble, Air Conditioning, and Refrigeration. Third Edition of part of Hirshfeld and Barnard's "Elements of Heat-Power Engineering," completely rewritten. 420 pages. 6 by 9. 209 figs.

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PART II

STEAM-GENERATING APPARATUS AND PRIME MOVERS, FUELS, COMBUSTION, AND HEAT TRANSMISSION

THIRD EDITION
OF SECOND PART OF
HIRSHFELD AND BARNARD'S
ELEMENTS OF HEAT-POWER ENGINEERING

COMPLETELY REWRITTEN
BY W. N. BARNARD AND F. O. ELLENWOOD

Corrected Printing, July, 1935

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NOTE REGARDING 1935 CORRECTED PRINTING

The need for a new printing of this volume has afforded opportunity for making typographical corrections, clarifying the text in a few places, and adding new references and a small amount of new material. The latter includes the data from the very complete tests of the new steam-generating unit in the Hudson Avenue Plant of the Brooklyn Edison Company (p. 456), a new table of data relating to large central stations in the United States (p. 140), and a low-pressure steam table (p. 786).

THE AUTHORS.

JULY, 1935.

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PREFACE TO PART II

This volume is intended primarily as a textbook covering the main apparatus in steam-power plants. Though written principally for college use, it is hoped that the book will also be valuable for reference in engineering offices and in libraries.

The work deals chiefly with the basic thermodynamics, structural and operating principles, and to some extent with the economic factors and elements of selection, as applied to steam prime movers and steam-generating apparatus; and it includes special chapters or sections devoted to the fundamentals of the following: steam turbines, the performance and determining features of prime movers, engine governors and valve gears, fuels, combustion, combustion apparatus for all types of fuels, interpretation of exit-gas analyses, heat transmission, furnaces, refractory and water-cooled furnace walls, boilers, superheaters, desuperheaters, reheaters, economizers, and air heaters.

The field covered is a somewhat extensive one, and in most cases the subjects considered in the various chapters would each require a large volume for exhaustive treatment; however, it is believed that the presentation herein is sufficiently complete to meet most requirements.

As in Part I, which is devoted mainly to the principles of thermodynamics, the arrangement and character of the presentation are such that, by the omission of less essential parts, the text can be used to advantage in short courses of instruction, the student at the same time obtaining a general conception of the broader scope of what he is studying and having incentive and means for further investigation. The numerous references given in each chapter will be of service to those desiring to pursue the subjects to a still greater extent. Many of the chapters have been written in such manner that they may be studied independently of the others, or in a sequence different from that followed in the book. Each chapter is accompanied by many questions and problems, and for most of the latter the numerical answers are given. Special effort has been made to prepare a comprehensive index which will be of real service to all users of the book.

In Part III, which is issued in a separate volume, there are treated the following additional subjects: flow of fluids through pipes, the draft system, the condenser system, feedwater heaters and allied equipment, boiler-water conditioning, properties of mixtures of air and water vapor, cooling ponds and towers, power plant ensemble, economics and cost data, air conditioning, and refrigeration.

The authors gratefully acknowledge their indebtedness to Professor Charles O. Mackey for preparing a large part of the chapter on heat transmission. Special credit is also due to Professor Robert P. Kolb and Mr. Howard N. Fairchild for valuable service extending over several months and to Professors Roy E. Clark and Warren H. Hook for assistance of various kinds.

Ithaca, N. Y.
August 10, 1933.

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HEAT-POWER ENGINEERING

PART II

CHAPTER XXI

STEAM TURBINES (*Continued*)

251. Introduction.—(**a**) In Chapter XX (Part I) steam turbines were discussed in a general way, hence it is assumed that the student is familiar with the main principles, types and features of prime movers of this kind, and also with their thermodynamic equations when considered as complete machines. It will be recalled that the ideal cycles that were developed in Chapters XVII and XVIII for all types of prime movers using vapors are applicable as standards with which to compare the actual performances of existing turbines, and that, by accounting for the probable losses, they may be used also for determining the results to be expected from proposed machines. The design of a new turbine is based on this estimated performance. Each of the various parts of the machine should, of course, be so proportioned that the performance of the turbine as a whole shall be the best obtainable under the limitations imposed by the strength of materials, the first cost, space occupied, reliability, and thermal economy of the machine.

(**b**) The present chapter will deal with some of the more important factors that enter into the thermodynamic calculations and into the design, selection, and arrangement of the various parts of turbines. But before proceeding with the considerations of these matters some may desire to make a brief review of the elements of the main types of turbines, and for that purpose Figs. 252 to 256 and Chart I are given.

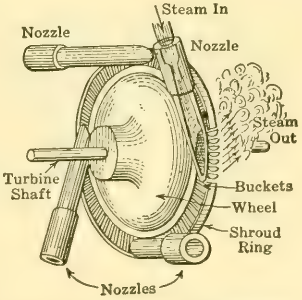
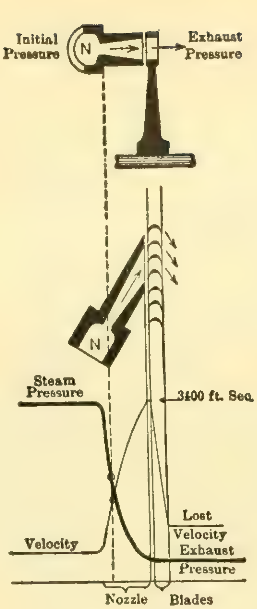


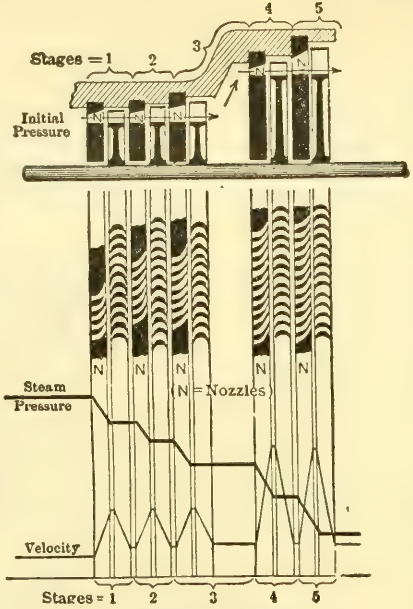
Fig. 252.—De Laval Elements.

The basic parts of a simple impulse turbine are shown in Fig. 252. Similar elements are shown in the second and succeeding stages of the

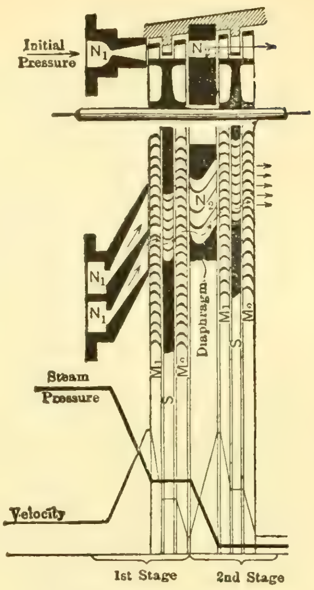
CHART I.—PRINCIPAL COMMERCIAL TYPES OF STEAM TURBINES.



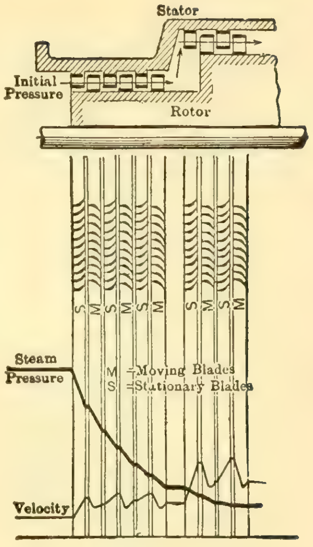
A Simple Impulse (De Laval)



B Pressure Staging (Rateau-Zoelly)



C Velocity Compounding (Curtis)



D "Reaction" (Parsons)

NOTES

General.—In each of the above diagrams the upper portion shows a longitudinal section of the turbine, the middle of the figure represents a transverse section through the nozzles and blades, and below this are curves which show how the pressure and velocity of the steam vary during the passage of the vapor through the turbine. The pressures and velocities are shown respectively by the ordinates of the heavy and of the light curves.

A. Simple Impulse (De Laval).—In this type of turbine it is to be particularly noted that full drop in pressure and the entire increase in velocity of the vapor are completed before the jet issues from the end of the nozzle, as shown by the curves; thus there is no expansion of the steam after it reaches the wheel casing. The velocity curve also shows the velocity-decrease of the jet resulting from the absorption of the kinetic energy by the buckets, and further shows the residual (lost) velocity due to the kinetic energy not utilized.

B. Pressure Staging-(Rateau-Zoelly).—With this form each pressure stage is seen to resemble a single-stage turbine of the De Laval type.

C. Velocity Compounding-(Curtis).—The diagram shows a turbine having two *pressure* stages, each of which has two *velocity* stages. It is seen by the curves that the pressure-drops and velocity-increases occur entirely within the nozzles N_1 and N_2 (i.e., there is no expansion of the vapor in the wheel casings). In each pressure stage the jet from the stationary nozzle first passes over the moving blades M_1 to which it surrenders part of its kinetic energy (thereby losing some of its velocity), and is then guided by the stationary blades S to act on the second set of moving blades M_2, which absorb still more of the energy by further decreasing the velocity of the jet. Thus the (kinetic) velocity energy is absorbed in two steps, or stages, in each pressure stage.

D. "Reaction"-(Parsons).—In this type, expansion takes place in *both* the stationary and the moving blades, as is shown by the steam-pressure line in the diagram. The steam is accelerated in passing through the first row of stationary blades; then, coming in contact with the first moving blades, it surrenders part of its kinetic energy, and while passing between these latter blades is further expanded and, finally, is discharged from them with a reaction. Thus the moving blades receive energy by both impulse and reaction This process is repeated in each of the succeeding pairs of stationary and moving sets of blades.

multi-stage impulse turbine illustrated in Fig. 253. The first stage of this turbine uses two-stage velocity compounding of the Curtis type, such as is shown in Fig. 254. The principal elements of rudimentary reaction turbines are given in Fig. 255; and Fig. 256 shows the elements of an early form of commercial multi-stage turbine of the so-called "reaction" type. Chart I supplements these figures and, by the curves thereon, shows how the pressures and velocities vary as the steam passes through the different parts of the machine. Turbine elements and arrangements are shown more fully in the latter part of this chapter.

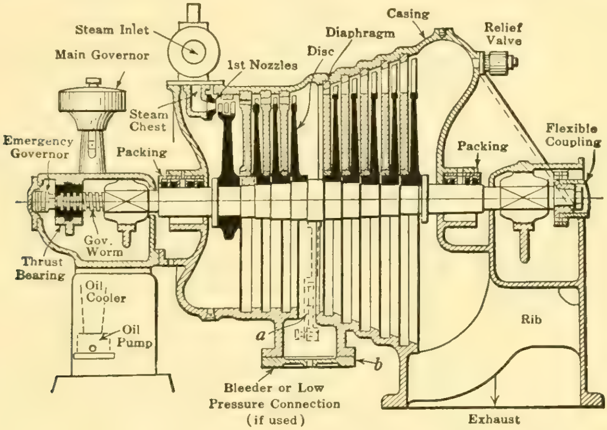


Fig. 253.—Elements of a Multi-stage Impulse Turbine.

(c) Since the thermodynamic process of transforming some of the heat of the working substance into the kinetic energy that is used to drive a turbine all takes place in the nozzles of the machine, and since the thermal performance of the turbine is largely dependent on the proper design of these elements, the principles of the flow through nozzles and the proportioning of these parts should be thoroughly understood. Furthermore, nozzles are not only used in steam turbines but also find application in many other devices, such as gas turbines, flow meters, feedwater injectors, and ejectors for removing non-condensable gases from condensers; and some of the principles involved in them are also useful in pro-

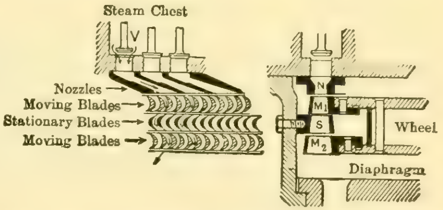


Fig. 254.—Velocity Compounding—Curtis Type.

portioning steam and gas piping and other passages through which the working substance flows.

The simplest case to analyze is that in which the working substance flows steadily and remains always in thermal equilibrium without losses of energy during the process. This ideal case will be considered first, and will be treated broadly in order to be applicable to both vapors and gases. Afterward, the modifications that are required to make this ideal case applicable to actual ones will be taken up. The cases in which the fluid becomes supersaturated and is present in two phases (e.g., is a mixture of vapor and moisture) during the process will also be considered.

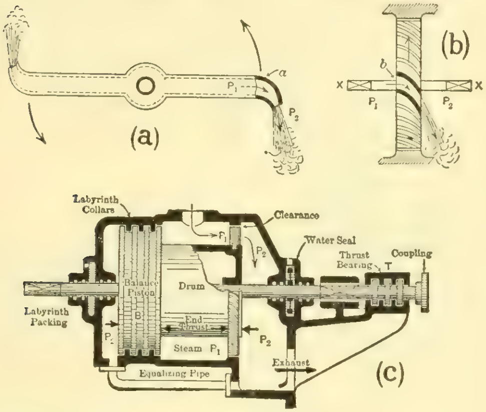


Fig. 255.—Simple Elements of a Reaction Turbine.

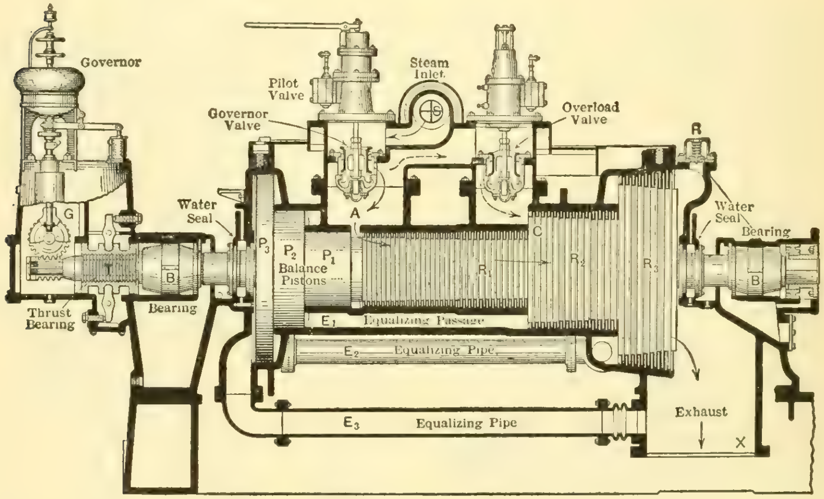


Fig. 256.—Elements of an Early Form of Reaction Turbine.

252. Flow of Fluids through Nozzles and Orifices.—(a) In all forms of steam and gas turbines the nozzle is the device used to produce the velocity energy of the working jet. Although some of the thermodynamic problems involved in a very careful analysis of the steam nozzle are rather complex and require considerably more study and research before any one can be certain as to the best methods and data to use in making nozzle calculations, the main problem can be treated in a general and comparatively simple way which brings out most of the important factors involved and which is sufficiently accurate for all applications except those requiring a rather high degree of refinement.[1]

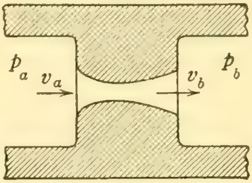


FIG. 257.

(**b**) Consider any fluid that is in thermal equilibrium while undergoing an adiabatic expansion in an ideal nozzle. Let this fluid be assumed to be passing through the stationary nozzle given in Fig. 257, the flow being produced by reason of the difference between the two constant pressures p_a and p_b maintained in the two vessels between which the nozzle is placed. By hypothesis this nozzle must be perfectly insulated so that the process can be adiabatic, and its shape must be such as to permit the steady flow of the fluid through it with thermal equilibrium of the fluid maintained. In other words, the fluid must pass through the ideal nozzle without any heat transfer, turbulence, friction, or supersaturation, and with perfect stream lines at all times. This also means that there is a uniform velocity across any transverse section of the fluid in the nozzle. Note, however, that this statement does not imply that the velocity at one section is equal to that at some other section. All of this information is conveyed in this text by saying the process is *isentropic.*

The **general energy equation** for steady flow, Eq. (22) developed in Chapter IV, Part I, being fundamental, applies of course to the case under consideration. When rewritten to conform to Fig. 257, this equation takes the form,

$$E_m' = H_a - H_b + E_h' + \frac{w(v_a^2 - v_b^2)}{50{,}000} + \frac{w(z_a - z_b)}{778} - R_c. \qquad (317)$$

in which E_m' = the mechanical energy delivered by the nozzle;

E_h' = the heat energy transferred to the fluid in the nozzle;

$H_a - H_b = w(h_a - h_b)$ = the *isentropic drop in the heat content of w pounds* of the fluid in passing through the ideal nozzle;

---

[1] Flow involving supersaturation and the presence of the fluid in two phases is briefly treated in Sect. 254.

R_c = the energy lost by radiation, conduction, and convection, as the fluid passes through the nozzle;

$z_a - z_b$ = the difference in elevation between the entering and the exit sections of the nozzle, and

$50{,}000 = 2g \times 778$.

For an **ideal nozzle**, R_c must be zero, and E_h' should also be taken as zero, because all actual nozzles are built without attempting to transfer heat to the fluid during its passage through them; E_m' must be zero because the sole function of the stationary nozzle is to increase the velocity of the fluid passing through it; and the term, $z_a—z_b$, is zero in a horizontal nozzle, and it approaches zero in the vertical one because this term involves an insignificant amount of energy compared with the other terms. Consequently, for the flow of a fluid through an ideal nozzle, Eq. (317) becomes

$$w(v_b^2 - v_a^2)/50{,}000 = H_a - H_b = w(h_a - h_b), \quad . \quad (318a)$$

or

$$v_b = \sqrt{50{,}000\,(h_a - h_b) + v_a^2}. \quad . \; . \; . \; . \; . \; . \; . \quad (318b)$$

If v_a is negligible relative to v_b, as often is the case, then

$$v_b = \sqrt{50{,}000\,(h_a - h_b)} = 223.7\sqrt{h_a - h_b}. \quad . \; . \quad (318c)$$

Using the constant 50,000 for $2g \times 778$ requires that v must be in feet per second and h in B.t.u. per pound.

(c) The **maximum jet energy** that can be acquired by any fluid in passing through an ideal nozzle under conditions of thermal equilibrium at all times, as specified in (**b**), is from Eq. (318), exactly equal to the isentropic drop in the heat content. This may also be referred to as the energy available from the ideal nozzle, or **the available energy.**

If this velocity energy could now be completely delivered to the shaft of a turbine running without any losses, the energy delivered by such a machine would be exactly represented, in ft-lb., by the area *labkl* in Fig. 258 for the reasons fully given in Sect. **201** (Part I). This area may therefore be said to represent also the jet energy acquired by passing through the ideal nozzle. In other words, this area represents the work done by any fluid in accelerating itself when passing through such a nozzle.

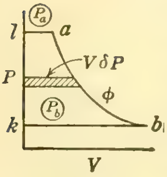


Fig. 258.

(**d**) If the fluid is now assumed to follow the law PV^n = a constant

(K) during its isentropic expansion from a to b, the work done in accelerating it, or the jet energy acquired by w pounds of this fluid, is

$$w\left(\frac{v_b^2 - v_a^2}{2g}\right) = \textit{area labkl} = -\int_{P_a}^{P_b} VdP, \text{ ft. lb.} \quad . \quad (319a)$$

in which the negative sign is necessary because the pressure is decreasing from a to b. But for the assumed fluid, since $PV^n = K$, it follows that

$$V = K^{\frac{1}{n}}P^{-\frac{1}{n}}$$

and

$$-\int_{P_a}^{P_b} VdP = -\int_{P_a}^{P_b} K^{\frac{1}{n}}P^{-\frac{1}{n}}dP = -K^{\frac{1}{n}}\left[\frac{P^{1-\frac{1}{n}}}{1-\frac{1}{n}}\right]_{P_a}^{P_b}$$

$$= -\left[\frac{P_b^{\frac{1}{n}}V_bP_b^{1-\frac{1}{n}} - P_a^{\frac{1}{n}}V_aP_a^{1-\frac{1}{n}}}{1-\frac{1}{n}}\right]$$

Hence,

$$w\left(\frac{v_b^2 - v_a^2}{2g}\right) = \frac{n}{n-1}[P_aV_a - P_bV_b] \text{ ft-lb.,} \quad . \quad . \quad . \quad (319b)$$

or, since $\frac{V_b}{V_a} = \left(\frac{P_a}{P_b}\right)^{\frac{1}{n}}$ for the assumed fluid, then

$$\frac{w(v_b^2 - v_a^2)}{2g} = \frac{n}{n-1}P_aV_a\left[1 - \left(\frac{P_b}{P_a}\right)^{\frac{n-1}{n}}\right] \text{ ft-lb.} \quad . \quad . \quad (319c)$$

In these equations it should be noted that $V_a(=w\overline{V}_a)$ and $V_b(=w\overline{V}_b)$ refer to the volume of the total weight of fluid that is being considered, and this weight w might be in pounds per hour or in pounds per second. If w is taken as one pound per second, for example, the values of V_a and V_b then become the specific volumes to be handled each second by the nozzle at the sections a and b, and will therefore be represented by $\overline{V}_a$ and $\overline{V}_b$ respectively.

(e) If the entering velocity v_a is so small relative to v_b that Eq.

(319c) is not materially affected by considering v_a to be zero, and if $w = 1$, then

$$\frac{v_b^2}{2g} = \frac{n}{n-1} P_a \overline{V}_a \left[1 - \left(\frac{P_b}{P_a} \right)^{\frac{n-1}{n}} \right], \quad . \quad . \quad . \quad . \quad . \quad (320a)$$

or

$$v_b = \sqrt{\frac{2gn}{n-1} P_a \overline{V}_a \left[1 - \left(\frac{P_b}{P_a} \right)^{\frac{n-1}{n}} \right]}. \quad . \quad . \quad . \quad . \quad (320b)$$

In this equation the discharge velocity v_b will involve the same units of time and distance that are used for $\sqrt{2g}$. For most purposes in this country it is best to use g in ft./sec.^{2}, in which case v_b will be in feet per second. $\overline{V}_a$ is the specific volume, in cubic feet per pound, of the fluid as it enters the nozzle. Then, since $P_a = 144 p_a$, it follows that, with v_b in feet per second,

$$v_b = \sqrt{64.34 \times 144 \frac{n}{n-1} p_a \overline{V}_a \left[1 - \left(\frac{P_b}{P_a} \right)^{\frac{n-1}{n}} \right]}$$

$$= 96.26 \sqrt{\frac{n}{n-1} p_a \overline{V}_a \left[1 - \left(\frac{p_b}{p_a} \right)^{\frac{n-1}{n}} \right]}. \quad . \quad . \quad . \quad . \quad . \quad (321a)$$

At any section x in this ideal nozzle at which the pressure has dropped to p_x, the velocity in feet per second is

$$v_x = 96.26 \sqrt{\frac{n}{n-1} p_a \overline{V}_a \left[1 - \left(\frac{p_x}{p_a} \right)^{\frac{n-1}{n}} \right]}, \quad . \quad . \quad . \quad . \quad (321b)$$

and, since the fluid in this nozzle is assumed to expand according to the law $PV^n = K$, its corresponding specific volume, in cu. ft. per lb., as it passes this section is

$$\overline{V}_x = \overline{V}_a \left(\frac{p_a}{p_x} \right)^{\frac{1}{n}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (322)$$

With w lb. of any fluid passing any section per second, the velocity, in feet per second, is v_x = (volume of flow, in cu. ft./sec.)/(area in sq. ft.) or

$$v_x = w \overline{V}_x / A_x. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (323)$$

The cross-sectional area, in square feet, at any section x is

$$A_x = w\overline{V}_x/v_x. \qquad (324)$$

Hence, for known values of p_a, $\overline{V}_a$ and n, substituting them and various values of the pressure p_x in Eqs. (321*b*) and (322), the corresponding velocities (v_x) and specific volumes ($\overline{V}_x$) are found; then, by using Eq. (324), the areas of the ideal nozzle are determined for as many sections as desired. When this is done it is found that, due to the simultaneous changes in velocity and specific volume, *the shape of the nozzle for gases and vapors is first convergent, and then, if the ratio of the discharge pressure to the admission pressure is low enough, the nozzle becomes divergent as indicated in Fig.* 259.

This shape results because at first the velocity increases more rapidly than does the specific volume, but further along the specific volume increases faster than the velocity. If the back pressure is sufficiently high, compared with the admission pressure, the nozzle will have only the contracting portion and is then called a *convergent nozzle.*[2]

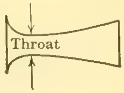


Fig. 259.

(f) The minimum cross-section of a convergent-divergent nozzle is called the **throat, or neck,** and the pressure and velocity at this point are called the **throat pressure** and **throat velocity,** or sometimes they are referred to as the **critical pressure** and **critical velocity** in the nozzle. For each set of conditions there is a definite relation between the critical pressure and the pressure at entrance to the nozzle.

The ratio of the throat pressure to admission pressure, or p_t/p_a, in the ideal nozzle will now be derived. Since each section of the nozzle has the same weight (w) of fluid passing through it in any unit of time, the minimum area or throat section must have a throat pressure (p_t) such that the resulting velocity and specific volume will give a greater *weight of flow per unit of area,* w/A, than any other section of the nozzle. In other words, w/A_x becomes a maximum at the neck, where $p_x = p_t$.

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[2] Such a nozzle, when short, is sometimes called an *orifice*, but the term *convergent nozzle* is probably superior. The distinction between short nozzles and orifices has not been well established in engineering literature, particularly as applied to steam; but the tendency seems to favor limiting the term "orifice" to openings that are very short relative to their diameters, such as those through thin plates. The flow of fluids through thin-plate orifices will be discussed later.

But from Eqs. (323), (321*b*) and (322), it follows that

$$\frac{w}{A_x} = \frac{v_x}{\overline{V}_x} = \frac{96.26\sqrt{\dfrac{n}{n-1}p_a\overline{V}_a\left[1-\left(\dfrac{p_x}{p_a}\right)^{\frac{n-1}{n}}\right]}}{\overline{V}_a\left(\dfrac{p_a}{p_x}\right)^{\frac{1}{n}}} \quad . \quad . \quad . \quad (a)$$

$$= \frac{96.26\,p_a^{\frac{n-2}{2n}}}{\overline{V}_a^{\frac{1}{2}}}\sqrt{\frac{n}{n-1}\left[p_x^{\frac{2}{n}} - \frac{p_x^{\frac{n+1}{n}}}{p_a^{\frac{n-1}{n}}}\right]}. \quad . \quad . \quad . \quad . \quad (b)$$

For w/A_x to be a maximum, dy/dp_x must equal zero, y being the bracketed quantity in Eq. (*b*). Thus,

$$\frac{dy}{dp_x} = \frac{2}{n}(p_x)^{\frac{2}{n}-1} - \frac{\dfrac{n+1}{n}(p_x)^{\frac{1}{n}}}{p_a^{\frac{n-1}{n}}} = 0, \quad . \quad . \quad . \quad . \quad (c)$$

or

$$\frac{2}{n} = \frac{\dfrac{n+1}{n}(p_x)^{1-\frac{1}{n}}}{(p_a)^{\frac{n-1}{n}}} = \frac{n+1}{n}\left(\frac{p_x}{p_a}\right)^{\frac{n-1}{n}}. \quad . \quad . \quad . \quad (d)$$

That is, at the neck where $p_x = p_t$,

$$\frac{p_t}{p_a} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (325)$$

Hence for any fluid that follows the law PV^n = *a constant* during its isentropic expansion in a nozzle, in which v_a is negligibly small, *the ratio of throat pressure to admission pressure* $\left(\frac{p_t}{p_a}\right)$ *is equal to* $\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$. Fig. 260 can be used for determining the numerical value of this ratio for any value of n within the usual range.

(g) The **critical velocity,** v_t, of any fluid expanding in a nozzle according to the law PV^n = *a constant* may now be found in terms of the exponent n, and of the pressure and volume either at the entrance or at the throat. Thus, from Eq. (325) it follows that

$$P_a = P_t \Big/ \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}},$$

and by hypothesis,

$$\overline{V}_a = \overline{V}_t\left(\frac{P_t}{P_a}\right)^{\frac{1}{n}}$$

$$= \overline{V}_t\left(\frac{2}{n+1}\right)^{\frac{1}{n-1}}.$$

Therefore,

$$P_a\overline{V}_a = \frac{n+1}{2}\,P_t\overline{V}_t.$$

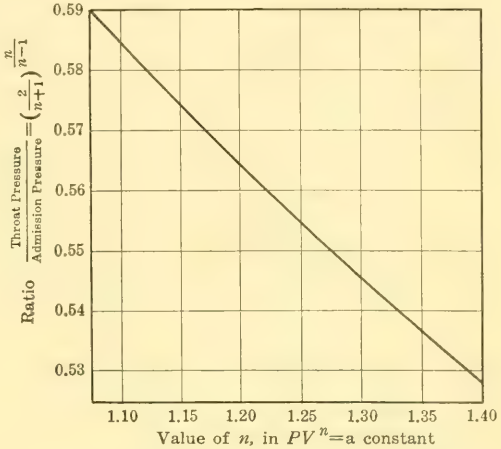

FIG. 260.—Ratio of Throat Pressure to Admission Pressure.

By combining this equation with 319(b), and assuming the entering velocity (v_a) to be negligible, the throat velocity becomes

$$v_t = \sqrt{2g\frac{n}{n-1}[P_a\overline{V}_a - P_t\overline{V}_t]}$$

$$= \sqrt{gnP_t\overline{V}_t}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (326a)$$

$$= \sqrt{2g\frac{n}{n+1}P_a\overline{V}_a}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (326b)$$

Equation (326a) expresses the fact [3] that the *critical velocity in a nozzle is equal to the velocity of sound in the given fluid when the pressure and specific volume of the latter are respectively* P_t and $\overline{V}_t$.

As an example of the application of the foregoing, consider air flowing through an orifice discharging against atmospheric pressure of 14.7 lb. per sq. in. and assume that the temperature of the issuing air is 60 deg. fahr. Then $P_t\overline{V}_t = RT_t = 53.3 \times 520 = 27{,}700$, $n = 1.4$, and $g = 32.17$.

Therefore, from Eq. (326a), $v_t = \sqrt{32.17 \times 1.4 \times 27{,}700} = 1{,}120$ ft. per sec. This value agrees well with the experimentally determined velocity of sound in air at this temperature and pressure.

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[3] See any good textbook on Physics.

From Fig. 260, for $n = 1.4$, it is found that $p_t/p_a = 0.528$, hence to have the throat pressure just equal to atmospheric, the admission pressure must be $p_a = 14.7/0.528 = 27.9$ lb. per sq. in. abs.

As another illustration, consider superheated steam to be issuing from a nozzle at a pressure of 14.7 lb. per sq. in. and with a temperature of 220 deg. fahr. The value of $\overline{V}_t$, as found from the steam tables or charts, would be 27.1 cu. ft. per lb., and the exponent n for the isentropic expansion of superheated steam at this pressure is known to be close to 1.3. Therefore for this case, the throat velocity is

$$v_t = \sqrt{32.17 \times 1.3 \times 144 \times 14.7 \times 27.1} = 1550 \text{ ft. per sec.}$$

The required steam pressure at admission for the exponent 1.3 will be found, by the aid of Fig. 260, equal to $14.7 \div 0.546 = 26.9$ lb. per sq. in. abs.

By means of steam charts and Eq. (318) this steam velocity may be checked very closely, showing that the value of n as chosen is a reasonable one for this case.

(h) The **values of the exponent n** depend on the kind of fluid flowing and its condition at entrance to the nozzle.

For *diatomic gases*, such as air, CO, N_2, and O_2, this exponent is equal to γ $(= c_p/c_v)$ which varies with the temperature from 1.40 at zero deg. fahr. to 1.23 at about 5000 deg., as shown in the Appendix in Plate VIII, which is Fig. 33 in Part I.

For *superheated steam* at pressures less than 500 lb. per sq. in. the exponent for isentropic expansion is close to 1.3, which gives a throat pressure of about 54.5 per cent of admission pressure.

For *saturated steam* in this same pressure region the exponent for equilibrium expansion in the nozzle is about 1.13, which means that the throat pressure is close to 58 per cent of the admission pressure.

If the value of the exponent n is not *very accurately* known for a given state of the fluid entering the nozzle, as in the case of vapors, the approximate value is useful in determining the throat pressure, p_t, by using the following cut-and-try method: First, substitute the approximate value of n in Eq. (325) and find a trial value of p_t. Next, by using a vapor chart that gives simultaneous values of specific heat content, pressure and entropy, find the value of the heat content (h_t) that results from isentropic expansion of the given fluid from the entering state to the trial value of p_t. The corresponding value of the trial throat velocity (v_t) may now be determined by using Eq. (318), and the corresponding specific volume may be calculated by using Eq. (322), or preferably read directly from the vapor chart. Consequently, for any known rate of flow, the area of the nozzle at the section having the trial pressure (p_t) may be obtained by using Eq. (324). By taking a few trial values of p_t, all fairly close together, the minimum cross-sectional area of the nozzle may thus be found.

(i) The **weight of fluid discharged** by a nozzle is fixed by the throat or smallest area of the nozzle, the throat velocity, and the specific volume of the fluid at that point. If the discharge pressure of a nozzle is *less* than the critical pressure, the divergent part of the nozzle becomes necessary in order to transform into jet-energy the additional energy made available by the lower discharge pressure. The *discharge velocity is increased by the divergent part of such a nozzle, but at the throat the velocity, specific volume, and area are not changed; consequently* **the weight of fluid discharged by a nozzle is not affected by a discharge pressure that is less than the critical.** *On the other hand, if the discharge pressure is greater than the critical, the weight discharged through a given minimum area is reduced*, and the nozzle will be of the *convergent* type.

These facts regarding the effect of the back pressure on the weight of fluid coming from a nozzle or orifice were first observed by Napier in 1867. However, Professor Osborne Reynolds, in 1866, was the first to show that the weight discharged is independent of the back pressure after the maximum velocity at the throat is reached, because this velocity is equal to that of sound in the given fluid at the throat condition *and no variation in pressure could be transmitted through the fluid at a greater velocity than this.* Many experiments since that time have verified this conclusion of Reynolds.

A *complete expansion nozzle* is one from which the fluid issues at a pressure just equal to that of the medium into which it discharges.

An **under-expanding nozzle** is one which discharges the fluid at a pressure greater than the external one, because the exit area is too small. In this case, the expansion is incomplete within the nozzle and, therefore continues after the issuance of the jet.

An **over-expanding nozzle** is one in which the area at exit is too large. With such a nozzle the pressure falls below the back pressure and then rises to the latter value at, or before, discharge.

The effect of either over-expansion or under-expansion in a steam nozzle is to reduce the exit velocity of the steam and therefore cause an energy loss. However, over-expansion is far more serious than under-expansion. Thus, if the exit area is about 20 per cent less than the area which just gives complete expansion, the velocity will be reduced by about 0.6 per cent, and there will be about a 1.2 per cent energy loss; whereas, if the exit area is 20 per cent too large, the loss in velocity will be about 4 per cent. and the energy loss will amount to approximately 8 per cent.

For this reason, some turbines, which operate most of the time under light loads, are equipped with nozzles that, at the rated load, have exit areas from 10 to 20 per cent too small to give complete expansion.

With this arrangement, underloads may be carried with throttled steam and yet maintain fair nozzle efficiencies. Another arrangement for such conditions is to have many nozzles and cut them in or out by hand- or automatic-controlled valves, so that for any load, insofar as possible, all of the nozzles in use will be completely expanding the steam, with consequent high efficiency.

(j) The **rational equations for finding the weight of fluid discharged** from ideal nozzles and orifices will now be derived.

Let w = weight of fluid discharged, in lb. per sec.;
A_t = area of nozzle throat, in sq. in.;
p_t = absolute pressure of the fluid at the throat, in lb. per sq. in.;
$\overline{V}_t$ = specific volume of the fluid at the throat in cu. ft. per lb. and
v_t = velocity of the fluid at the throat in ft. per sec.

Assuming that the *discharge pressure* (p_b) *is less than the critical*, that the fluid enters the nozzle in the state a with a negligible velocity, and that it follows the law PV^n = *a constant* during its passage through the nozzle, then, from Eq. (321a), the throat velocity becomes

$$v_t = 96.26\sqrt{\frac{n}{n-1}p_a\overline{V}_a\left[1 - \left(\frac{p_t}{p_a}\right)^{\frac{n-1}{n}}\right]}.$$

But

$$w = \frac{A_t v_t}{144\overline{V}_t}, \quad \text{and} \quad \overline{V}_t = \overline{V}_a\left(\frac{p_a}{p_t}\right)^{\frac{1}{n}}.$$

Therefore,

$$w = \frac{96.26A_t}{144\overline{V}_t}\sqrt{\frac{n}{n-1}p_a\overline{V}_a\left[1 - \left(\frac{p_t}{p_a}\right)^{\frac{n-1}{n}}\right]}$$

$$= 0.668A_t\sqrt{\frac{n}{n-1}\frac{p_a}{\overline{V}_a}\left[\left(\frac{p_t}{p_a}\right)^{\frac{2}{n}} - \left(\frac{p_t}{p_a}\right)^{\frac{n+1}{n}}\right]}. \quad . \quad . \quad (327)$$

If the fluid is one that also follows the law $P\overline{V} = RT$, then, with the entering absolute temperature T_a,

$$w = \frac{8.02p_aA_t}{\sqrt{RT_a}}\sqrt{\frac{n}{n-1}\left[\left(\frac{p_t}{p_a}\right)^{\frac{2}{n}} - \left(\frac{p_t}{p_a}\right)^{\frac{n+1}{n}}\right]}. \quad . \quad . \quad . \quad (328)$$

For *air at ordinary temperatures* the specific heats may be considered as constant, with $n = 1.4$ and $R = 53.3$. With this value of n used in Eq. (325) the critical pressure for air becomes $0.528\,p_a$. Consequently the maximum weight of air from a nozzle with a discharge pressure less than the critical is obtained by substituting these values in Eq. (328), giving

$$w_{air} = 0.53 p_a A_t \sqrt{T_a}. \qquad (329)$$

Similarly, for *superheated steam*, at pressures of 500 lb. per sq. in., or less, n is very close to 1.3 and the critical pressure thus becomes 0.546 p_a. Hence the maximum discharge of superheated steam for such pressures becomes

$$w_{sup.\ steam} = 0.316 A_t \sqrt{p_a / \overline{V}_a}. \qquad (330)$$

For *saturated steam* the value of n varies from about 1.14 at low pressures (below atmospheric) to about 1.12 at pressures around 500 lb. per sq. in. Using $n = 1.13$, the critical pressure becomes 58 per cent of the admission pressure, and thus the maximum weight of initially saturated steam at pressures less than 500 lb. per sq. in. that would flow from a nozzle having uniform velocity of vapor and moisture would be

$$w_{sat.\ steam} = 0.3 A_t \sqrt{p_a / \overline{V}_a}. \qquad (331)$$

The reason for stipulating that this equation is based on uniform velocity of the vapor and moisture will appear in the discussion of two-phase velocities in Sect. **254.**

For steam pressures above 500 lb. per sq. in., the value of n is reduced appreciably below the values given above. For instance, with superheated steam expanding from $p = 2000$ to $p = 500$, $n = 1.25\pm$, according to calculations based on Keenan's Steam Tables.

When the *discharge pressure is greater than the critical*, equations (327) and (328) may also be used by substituting in them the known discharge pressure, p_b, for p_t. *For vapors*, the determination of the weight discharged is usually made by finding, by the aid of suitable vapor charts, the velocity and the specific volumes at the end of isentropic expansion because this method is more accurate and simpler than using the equation $PV^n = K$. Nevertheless the equations for steam with discharge pressures greater or less than the critical are often needed for reference and they are therefore given in convenient form in Table XV.

The *actual* weight of air or superheated vapor discharged from any real nozzle is slightly less than from the ideal, as given by the equations in this table, because the actual flow is not quite isentropic. For an *orifice* that does not have a well-rounded entrance there may be a still greater difference because there will also be considerable contraction of the issuing jet. The velocity and discharge coefficients of nozzles

TABLE XV

Equations for Finding the Maximum Velocity and Weight Discharged from Ideal Nozzles and Orifices

For steam at pressures above 500 lb. per sq. in. the table should not be used as the exponents then become appreciably less than those given.

For coefficients of velocity and discharge for various cases, see Sects. **253** and **254**.

Fluid	Exponent for Isentropic Expansion	Ratio $\left\{\frac{\text{Back Pressure}}{\text{Admission Pressure}}\right\}$	Weight Discharged from an Ideal Nozzle or Orifice Having a Throat Area of A_t Square Inches and Uniform Velocity of the Jet, Pounds per Second w	Velocity of Discharge from Ideal Nozzle or Orifice, in Feet per Second v_b
Air with $t_a < 500°$ fahr.	1.4	$\frac{p_b}{p_a} \eqless 0.53$	$\frac{0.53 p_a A_t}{\sqrt{T_a}}$	$109.5\sqrt{T_a\left[1-\left(\frac{p_b}{p_a}\right)^{0.286}\right]}$ $(=44.6\sqrt{T_a}$ for the critical velocity)
		$\frac{p_b}{p_a} > 0.53$	$\frac{2.05 p_a A_t}{\sqrt{T_a}}\sqrt{\left(\frac{p_b}{p_a}\right)^{1.43}-\left(\frac{p_b}{p_a}\right)^{1.71}}$	
Superheated steam, $p_a < 500$ lb. per sq. in. . .	1.3	$\frac{p_b}{p_a} \eqless 0.55$	$0.316 A_t\sqrt{\frac{p_a}{\bar{V}_a}}$	$200\sqrt{p_a\bar{V}_a\left[1-\left(\frac{p_b}{p_a}\right)^{0.231}\right]}$ $(=72\sqrt{p_a\bar{V}_a}$ for the critical velocity)
		$\frac{p_b}{p_a} > 0.55$	$1.39 A_t\sqrt{\frac{p_a}{\bar{V}_a}\left[\left(\frac{p_b}{p_a}\right)^{1.54}-\left(\frac{p_b}{p_a}\right)^{1.77}\right]}$	
Saturated steam, $p_a < 500$ lb. per sq. in. . .	1.13	$\frac{p_b}{p_a} \eqless 0.58$	$0.3 A_t\sqrt{\frac{p_a}{\bar{V}_a}}$	$284\sqrt{p_a\bar{V}_a\left[1-\left(\frac{p_b}{p_a}\right)^{0.115}\right]}$ $(=70\sqrt{p_a\bar{V}_a}$ for the critical velocity)
		$\frac{p_b}{p_a} > 0.58$	$1.97 A_t\sqrt{\frac{p_a}{\bar{V}_a}\left[\left(\frac{p_b}{p_a}\right)^{1.77}-\left(\frac{p_b}{p_a}\right)^{1.89}\right]}$	

and orifices are given in Sect. **253** for air and superheated steam. Sect. **254** deals with the more complex factors involved in the flow of *wet* steam.

(k) **Empirical formulas for finding the weight of steam discharged by an orifice** will now be given in order that they may be conveniently compared with the rational equations of Table XV. Sometimes these empirical equations are sufficiently accurate for the purpose needed and their simplicity recommends them in such cases.

Let p_a = absolute admission pressure to the orifice in lb. per sq. in.;
p_b = absolute back pressure on the orifice, in lb. per sq. in.;
A = area of the orifice in sq. in., and
w = weight discharged in lb. per sec.

Then for *saturated steam*, if p_b is less than the critical, by **Grashof's formula,**

$$w = 0.0165 A p_a^{0.97}, \quad \ldots\ldots \quad (332)$$

by **Rateau's formula**

$$w = A p_a\,(16.367 - 0.96 \log p_a)/1000 \quad (333)$$

and by **Napier's formula**

$$w = p_a A/70. \quad \ldots\ldots\ldots \quad (334)$$

Grashof's formula is sometimes modified, by the use of proper coefficients, so as to be applicable to the flow of wet or superheated steam. If the entering steam is *wet*, the discharge calculated as above is to be divided by $\sqrt{x_a}$, where x_a is the quality of the steam at entrance to the orifice. If the entering steam is *superheated*, the weight of steam discharged, calculated from the preceding formula, is to be divided by $(1 + 0.00065D)$, where D is the degree of superheat of the steam in the reservoir.

When $\frac{p_b}{p_a}$ is greater than the critical ratio the *rationally derived equations* must be used. Such equations always give more nearly correct results than the empirical ones and should always be applied whenever accuracy is important.

253. Nozzle and Orifice Coefficients for Gases and Vapors. (a) The **velocity coefficient** is the ratio of the actual discharge velocity to the corresponding ideal velocity. But the application of this definition requires very careful consideration because the *true average* velocity across a section of the issuing jet may not be exactly the same as that obtained by some of the experimental methods used and because the material may not be homogeneous, as is the case when the fluid is a mixture of moisture and vapor.

(b) The *average velocity that engineers need to use* in turbine calculations is the one that is measured in terms of the impulse (action) or reaction of the jet of the actual nozzle, as these are the forces that drive the turbine. These forces can be determined by test. But in the *actual* nozzle, the velocity is not uniform over the cross-section of the nozzle and the working substance may not be homogeneous. Hence the mean jet velocity needed in the discussion of velocity coefficients is the velocity of a jet having uniform rate of flow across its transverse section, which would give a mass flow and momentum in a unit of time equal respectively to the mass flow and momentum in the same unit of time of the actual jet. The ratio of this mean velocity to that from the ideal nozzle for similar conditions gives the *desired velocity coefficient*, which will be designated by c. Thus the **velocity coefficient of a nozzle** needed by turbine engineers, is

$$c = \frac{\text{actual discharge velocity as measured by impulse or reaction}}{\text{corresponding ideal velocity, from Eq. (318)}}. \qquad (335)$$

For all cases, the *corresponding ideal velocity*[4] is the one that would be attained by a homogeneous fluid in passing through a frictionless nozzle without turbulence or heat transfer—the state of the fluid entering the ideal nozzle, and also the discharge pressure, being the same as in the actual nozzle. Note that this similarity is not always sufficiently attained by having the pressure and temperature of the steam entering the ideal nozzle the same as in the actual one, but in addition the entrance velocity must also be the same, when any appreciable energy is involved in this term.

(c) The *actual variation of steam jet velocities* over the cross-section of a nozzle is very difficult to obtain. However, observations by Stodola,[5] who has experimentally determined this variation, show that the velocity is a maximum at the axis of the nozzle, but remains substantially constant over about four-fifths of the cross-section, and then drops very rapidly in the outer section, approaching zero at the periphery. In fact, his experiments show that in the outermost section of a width of about one-twentieth of the radius the average velocity is approximately one-half of the maximum velocity at the axis.

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[4] The term "*theoretical velocity*" is sometimes used instead of "*ideal velocity*," but the former is far less satisfactory because nothing is implied by the name as to how much "theory" is included. It is perfectly logical, although very difficult, to calculate a theoretical velocity that includes the effect of friction, turbulence, and the variation in velocity at any cross-section. The name *ideal velocity*, however, naturally carries with it the idea of the elimination of all nozzle losses.

[5] "Steam and Gas Turbines," by Stodola, p. 66, Vol. I.

(d) **The values of the velocity coefficients** of well-designed nozzles used in steam turbines generally lie between 95 and 99 per cent for superheated steam, but with wet steam they are appreciably less, as shown in Sect. **254**(**h**). The losses up to the throat of a nozzle are often negligible.

Accurate testing of steam nozzles for determining velocity coefficients is a very difficult task and the experimental data thus far available show considerable variation in their results. The chief reasons, probably, for most of these variations are:

(1) Lack of agreement on the thermal properties of steam.
(2) Lack of knowledge, at the time the tests were made, of the importance of many factors that are now known to affect the results.
(3) Inaccurate data obtained by reason of imperfections in the apparatus used in the tests.

From the knowledge now available it is known that a reduction in the velocity coefficient is caused by having:

(1) Rough surfaces.
(2) Moisture.
(3) Improperly rounded entrance.
(4) Improper divergence for discharge pressures less than the critical.
(5) Large degree of curvature, especially with short radius.
(6) Very small nozzles.
(7) Long nozzles.
(8) Thick division plates between parallel nozzles.

There is still considerable uncertainty regarding the effect of pressure, temperature, and velocity on the velocity coefficients. The best data indicate that the coefficients are not appreciably affected by velocities ranging from about 400 to 2000 ft. per sec., but above the latter value there is a rapid reduction in the coefficients.[6]

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[6] Further discussion of this subject may be found in good textbooks on steam turbines, and the following references are especially recommended to those desiring more complete information:

A Machine for Testing Steam Turbine Nozzles by the Reaction Method, by Warren and Keenan, Trans. A.S.M.E., 1926, p. 33.

Reports of the Steam Nozzles Research Committee, Institution of Mechanical Engineers, Proceedings 1923, 1924, 1925, 1928, and 1930. (The Sixth and final report is abstracted in Engineering, Feb. 28 and March 14, 1930.)

(e) The **discharge coefficient** (c_d), or the **flow coefficient** as it is sometimes called, represents the ratio of the *weight* actually discharged in a unit of time to the weight that would be discharged in the same time by an ideal nozzle, or orifice, having no contraction of the jet and also having uniform velocity of the jet at any cross-section of it. The coefficient of discharge is affected by the following factors:

(1) The contraction of the jet (especially important in the square-edged orifice).
(2) The ratio of the actual average specific volume of the fluid at the throat to that calculated for isentropic expansion.
(3) The ratio of the actual average throat velocity to that calculated for isentropic expansion. (This is especially important in the flow of a wet vapor and is discussed under two-phase velocities in the next section.)
(4) The uncertainties that may be involved in the equations used to find the throat velocity and specific volume of the given fluid in the ideal nozzle.
(5) The errors that may be involved in the measurements of the actual pressures, temperatures and weight of fluid discharged during the test.

A nozzle may have a high discharge coefficient yet may be unsuitable for use in a turbine because the stream lines of the issuing jet lack the parallelism necessary to enable them to act efficiently on the blades of the turbine rotor. The velocity coefficient, however, is a true measure of the value of the nozzle for such applications, because its magnitude is dependent on the parallelism of the stream lines with respect to the axis of discharge. The actual discharge from a steam nozzle can be determined easily by condensing the issuing steam and weighing it.

(f) The **values of the discharge coefficients** for well-designed nozzles **for steam**[7] ordinarily range from 0.97 to 1.00; but when there is supersaturation and two-phase flow, which are considered in Sect. **254,** the apparent discharge coefficient may even be greater than unity.

(g) For **air at ordinary atmospheric temperature** and with pressures ranging from 70 to 25 lb. per sq. in. abs., the results of Morley[8] show discharge coefficients from 95 to 98 per cent for nozzles with rounded entrances, throat diameters of about 0.2 in., and lengths varying from about 0.2 in. to nearly 3 in. His experiments show that the long nozzle with well-rounded entrance is superior to the short one with a small radius of curvature at entrance.

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[7] See references in footnote 6.

[8] Flow of Air through Nozzles, Inst. Mech. Eng'rs., Jan. 1916, p. 51.

For **air and other gases at high temperature** the experimental results are naturally much more meager than for ordinary temperatures. However a limited amount of information on $\frac{1}{4}$ in. diameter nozzles 1 in. long with well-rounded entrances, for gage pressures ranging from 30 to 100 lb. per sq. in. and absolute temperatures from 300 to 1400 deg. cent., is given by E. C. Wadlow.[9] These coefficients are based on calculated discharges using the *variable* specific heats of gases, as given by Partington and Shilling. The gases actually used were the products of combustion of Mond gas and air and their composition varied considerably during the tests. The results indicate that for convergent-parallel nozzles the discharge coefficients increase on the average from a value of about 85 per cent at 300 deg. cent. abs. to about 90 per cent at 1400 deg. cent. abs., with a variation of 4 per cent on either side of these figures; whereas, with the convergent-divergent nozzle, the coefficients are about 2 per cent higher.

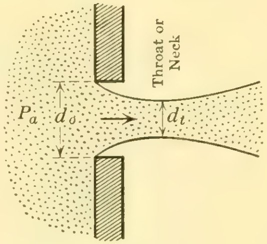

FIG. 261.

(**h**) For **thin-plate orifices** the *velocity coefficients* are about the same as for nozzles, but for a square-edged orifice the *discharge coefficients* are materially reduced due to the contraction of the jet, as shown by Fig. 261. Many tests have been made on this type of orifice but, generally, the pressure ratio has covered only a limited range. This ratio is clearly an important factor in the discharge coefficient as shown by Fig. 262.

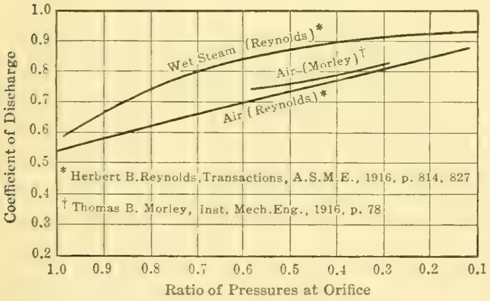

FIG. 262.—Discharge Coefficients for Thin-Plate Orifices.

(i) The **thin-plate orifice as a fluid meter** is very extensively applied in measuring the rate of flow of gases and vapors. When used for this purpose the orifice is inserted in a pipe line and the area of the orifice is an appreciable portion of that of the pipe. Consequently the drop in pressure due to the orifice is so small that it is usually measured by some form of differential pressure gage. This type of meter is not recommended by the American Society of Mechanical Engineers [10] for rates of flow which make the pressure differential more than 1.5 per cent of the absolute pressure before the orifice, unless it can be standardized by a special test. With the thin-plate orifice used within this 1.5 per cent range the coefficients of discharge that should be used are given by Fig. 263, which is reproduced from the A.S.M.E. reference just given. From this figure it is evident how important are the orifice ratio and the location of the pressure taps on the discharge side of the orifice.

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[9] The Expansion of High-Temperature Gases in Nozzles, Inst. Mech. Eng'rs., Vol. I, 1928, p. 405.

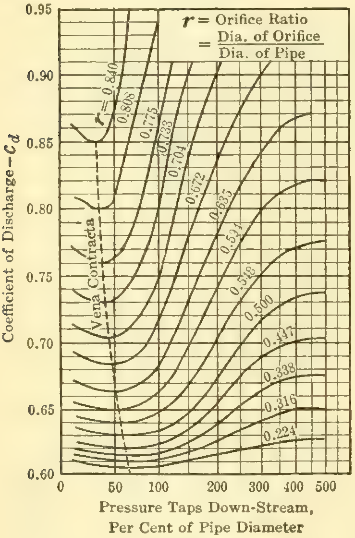

FIG. 263.

To those who are specially interested in the subject, this Report on Fluid Meters is recommended because it gives a comprehensive discussion of the various types of meters and an excellent bibliography. Bulletin 207, Univ. Illinois Eng. Exp. Station, May, 1930, by Polson, Lowther, and Wilson, on *The Flow of Air through Circular Orifices with Rounded Approach,* gives experimental data for pressures from 1 in. of water to 35 in. of mercury, and also gives an extensive bibliography. Also see Research Paper 49, Bureau of Standards, 1929, by Bean, Buckingham and Murphy on Discharge Coefficients of Square-edged Orifices for Measuring the Flow of Air.

254. Actual Nozzle Discharge of Wet Steam; Supersaturation and Two-phase Velocities.[11] (a) In the nozzle equations previously derived, all of the fluid passing any section at any instant of time has been assumed to be in thermal equilibrium. In the strict sense, this means not only a uniform temperature and pressure throughout the fluid at any section, but also a uniform velocity of all parts of the fluid at this section. This condition therefore eliminates all friction and turbulence, and consequently there can be no lagging behind of liquid particles that might exist in the mixture at entrance to the nozzle, or that might possibly be produced by the expansion of the fluid within the nozzle. *For superheated steam* this assumption is entirely satisfactory, as many tests of actual nozzles using such steam have shown the discharge to be less than that from the corresponding ideal nozzle, as would be expected. This difference is easily accounted for by applying a coefficient slightly less than unity to account for the fluid friction and turbulence that must always be present, to some degree at least, in any real nozzle.

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[10] See p. 39, Fluid Meters, Part I, Second Edition, A.S.M.E. Report of Special Research Committee, 1927.

[11] This section may be omitted in short courses; but for those desiring additional information, see "Supersaturated Steam" by John I. Yellott, Jr. Tr. A.S.M.E., June, 1934, p. 411.

On the other hand, tests of actual nozzles receiving *wet steam*, or even a few degrees of superheat, show that the actual discharge *is often greater* than the value calculated for the ideal nozzle, as developed in Sect. **252.** This discrepancy increases as the percentage of liquid initially present becomes greater, until, for example, the measured discharge becomes from 6 to 8 per cent greater than the calculated value with steam containing 15 per cent moisture at entrance to the nozzle. This anomaly has long puzzled engineers because the uncertainties regarding the correct values of the properties of steam are too slight to account for such a large difference between the calculated and the measured discharge.

(**b**) At the present time it appears, to the authors at least, that the matter is likely to be cleared up by the consideration of **supersaturation** combined with what may be called **"two-phase velocities."** The latter term simply means that two different velocities may be attained by the two components of any two-phase substance, such as a mixture of steam and entrained drops of liquid, when this substance passes through a nozzle or pipe line. So far as the authors are aware, the effect of an appreciable difference in the velocities of the two phases of a fluid in a nozzle has not been carefully considered until very recently. The effect of supersaturation and two-phase velocities can only be discussed here in brief form, but references will be given so that further study may easily be pursued by those desiring it. In this connection note that the flow of *wet* steam through a nozzle is not of the commercial importance today that it was a few years ago, because in nearly all modern turbines the steam is in a superheated state at the entrance to the nozzles in most stages. Even in the last stages the steam may be dry or substantially so, if reheating is used. However, there are other cases in which nozzles and orifices may be used to measure the flow of wet vapor, and this fact, combined with the importance of the scientific principles involved, justifies some consideration of this flow, aside from its possible use in turbine calculations.

(**c**) As long ago as 1870 it was first shown by Lord Kelvin [12] that a mixture of a vapor and very fine drops of its liquid cannot be in equilibrium unless the vapor pressure exceeds the normal saturation pressure by a certain amount that depends upon the curvature of the surface of the liquid, its temperature, and the nature of the liquid. The surface tension of small drops of a liquid involves an appreciable amount of energy, so that the energy contained in a given weight of liquid at any specified pressure and temperature is greater when in the form of minute drops than when in the form of a united body of liquid. The condition for equilibrium between a vapor and minute drops of its liquid will now be considered.

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[12] Edinburgh Royal Society Proceedings, Vol. 7, p. 63.

If a vessel, as A in Fig. 264, contains nothing but steam above a *flat* surface of water, the mixture in this vessel is said to be in *equilibrium* whenever the rate of evaporation of this water is just counterbalanced by the rate of condensation of the steam. Molecules are constantly passing from the surface of this fluid into the vapor, and an equal number are being condensed in the same unit of time. The vapor pressure just above this flat liquid surface is the so-called *saturation pressure*, or better the *normal saturation pressure*, of the vapor corresponding to its temperature. Now suppose that a very fine capillary tube B, that *is not wet by the water*, is attached to the vessel A; and consider this tube to be partially filled with water, above which there is nothing but steam. The capillary action between the dry tube and the water will cause the liquid to have a convex surface as indicated in the figure. The finer the tube the smaller becomes the radius of curvature of this surface, and the greater is the depression d of this fluid in the tube when connected to vessel A at the top and bottom. Consequently the pressure of the vapor just above this convex liquid surface in B will be greater than it is in A, or greater than the normal saturation pressure, by a certain amount due to the extra weight of the column of vapor of depth d, and the vapor is said to be **supersaturated.** For the liquid just under the curved surface the entire pressure from above is equal to (pressure of the supersaturated vapor on the curved surface) + (pressure due to surface tension of the curved surface), and an exactly equivalent total pressure from below is due to (pressure of the saturated vapor on the flat surface) + (pressure due to the liquid of depth d). This system will be in equilibrium when the temperature is the same throughout, because the rate of evaporation from each liquid surface in A and B is just balanced by the rate of condensation thereon.

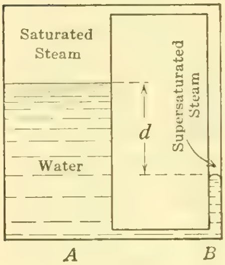

Fig. 264.

(d) Again, consider **fine drops of a liquid to be entrained in an atmos-**

phere of its own vapor, *all at the same temperature. This vapor must have a higher pressure than its normal saturation pressure, corresponding to this temperature, in order to be in equilibrium with all these curved surfaces of the liquid,* just as was the case with the liquid in the capillary tube. In other words, for such a system to be in equilibrium it is necessary that this vapor shall be *supersaturated,* which means that its pressure is higher than the normal saturation pressure corresponding to its temperature. The **degree of supersaturation** of a vapor is defined by the ratio,

$$\frac{\text{Actual pressure of vapor}}{\text{Normal saturation pressure corresponding to the temperature}}.$$

Consequently for any temperature there is only one degree of supersaturation of a vapor that will enable it to be in equilibrium with the millions of convex curved surfaces of its liquid when in the form of small drops of any given diameter. The supersaturated vapor is sometimes called **undercooled vapor** because its temperature is less than the normal saturation temperature corresponding to its actual pressure.

(**e**) The relation between the diameter of drop, the temperature, and the degree of supersaturation, of steam for equilibrium is expressed by the **Kelvin-Helmholtz formula,**[13] which is given in English units as follows:

$$\log_e \frac{p}{p_s} = \frac{2S}{\rho\, rRT}, \qquad (336)$$

where ρ = density of liquid, in lb. per cu. ft.;
p = pressure of the supersaturated vapor;
p_s = saturated pressure of vapor at the known temperature;
r = radius of the liquid drop, in feet;
R = the gas constant for the vapor in $PV = wRT$;
= 85.8± for steam;
S = surface tension of the liquid, in lb. per ft.;
T = absolute temperature, in deg. fahr.

To show the rapid increase in the degree of supersaturation with the decrease in the radius of very small drops (such as one ten-millionth of an inch, or about 10^{-8} ft.) the following values from Eq. (336) for $T = 700$ deg., as calculated by Goodenough,[14] are given:

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[13] For the derivation of this formula, see Stodola, Steam and Gas Turbines, p. 1035; also Ewing, Thermodynamics for Engineers, p. 347.

[14] Power, p. 468, Sept. 27, 1927.

Radius of drop, in ft., $r = 10^{-5}, \quad 10^{-6}, \quad 10^{-7}, \quad 10^{-8}, 10^{-9}$

Degree of supersaturation, $\frac{p}{p_s} = 1.0002, \ 1.0021, \ 1.021, \ 1.23, \ 8.15$

This table shows that a "micro-drop" of water surrounded by steam, all at a temperature of 240 deg. fahr., for which the saturation pressure is 24.97 lb. per sq. in., will require a vapor pressure of 1.23 × 24.97 or 30.7 lb. per sq. in. to be in equilibrium if the radius of the drop is 10^{-8} ft. If the drop is smaller, it will evaporate unless it is surrounded by steam of a higher degree of supersaturation; and conversely, a larger drop will continue to increase by condensation of vapor upon it until the degree of supersaturation is reduced sufficiently to approach equilibrium. When the radius becomes as large as 10^{-5} ft., the vapor at the given temperature is then almost saturated as shown by the ratio 1.0002 in the table. On the other hand, the smallest drop that can exist is one consisting of a single molecule of water, which means a radius not much less than 10^{-9} feet, or approximately the one hundred-millionth part of an inch.

(f) **When superheated steam expands** so that it crosses the saturation curve, the development of relatively large drops of liquid is necessary if this fluid is to become a mixture of a saturated liquid and vapor. But the formation of such drops requires a small period of time (the exact value being unknown) and the experiments of Aitken,[15] Wilson,[16] and Stodola[17] all show that there must be small dust particles or ions, to act as condensation "nuclei" on which the drops may begin to form. If there are only a few of these nuclei present the expansion will proceed much further before condensation begins than when the nuclei are numerous. Consequently dry steam that is relatively free of dust or electrically charged molecules will expand quickly in a nozzle to a high degree of supersaturation before condensation begins. There are then formed, as determined by Yellott, very minute particles of liquid which do not grow. Further condensation appears to occur by the formation of new droplets rather than by the growth of those which have previously been formed. If the steam entering the nozzle is wet, condensation does not take place on the existing particles, because they have a higher temperature than the surrounding steam; instead, new droplets are formed just as if the entering steam had been dry.

The time required for saturated equilibrium conditions to be attained by steam under different conditions in steam turbines is not definitely known; but experiments show that for a given temperature there is a definite degree of supersaturation at which condensation occurs and thermal equilibrium is established to agree with the properties of wet steam. This limit of the supersaturated state has been called the "Wilson line" by Martin,[18] and Yellott finds that it corresponds to about 3.5 per cent moisture.

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[15] Edinburgh Royal Society, Trans., Vol. 30, 1883, p. 337.

[16] Philosophical Trans. of the Royal Soc. of London, 1897, Vol. 189, p. 265.

[17] Steam and Gas Turbines, Vol. I, p. 119.

Regardless of whether or not equilibrium is ever attained in a turbine after supersaturated or undercooled steam is produced in a nozzle, consider how this steam affects the weight discharged from the nozzle. When *superheated* steam, at ordinary pressures, expands adiabatically under equilibrium conditions, the resulting pressure-volume relation may be rather closely expressed by the equation $PV^{1.3}$ = a constant. When such steam expands into the region below the saturation curve without any condensation, this same equation probably holds fairly well. On the other hand, the adiabatic expansion of *saturated* steam under equilibrium conditions gives an equation that may be approximated by the expression $PV^{1.13}$ = a constant. These exponents affect in two ways the weight discharged by a nozzle. The larger the exponent the smaller becomes the velocity and also the specific volume for any specified ratio of pressures involved in the expansion. Therefore the specific volume and velocity of the undercooled steam are both appreciably less than those of the saturated steam for the same entering state and the same exit pressure, but they are not affected in the same proportion. The reduction in the specific volume of the undercooled steam is more than sufficient to offset its decreased velocity, so that *a nozzle will discharge a greater weight of supersaturated steam than of the saturated*, the difference often being as much as 2 per cent.

(g) **Two-phase velocities** will now be considered, because the discharge of wet steam from a nozzle is greatly affected thereby. Whenever a vapor or gas moving at high velocity, as in a pipe line or nozzle, has drops of liquid entrained in it, the size of drops that can be carried is limited by the relation existing between the viscosity of the gas, the densities of the liquid and gas, and their relative velocities. The surface tension of the liquid tends to hold the drop together and the rush of gas past the drop tends to tear it apart. For a full discussion of this relation, see articles by Upton [19] and Glazebrook.[20]

After water drops have been formed in a steam nozzle, or if they existed at entrance to the nozzle, they will soon be enveloped by steam moving faster than the drops because the drops of water can not give up much energy to produce their own acceleration, compared with the energy available from the expanding vapor to accelerate itself. That a saturated liquid flowing through a nozzle is not self-accelerating to any degree comparable with an equal weight of saturated vapor at the same temperature may be readily seen by comparing the two widely different values of the isentropic drop in heat content, computed for the same pressure limits of the two distinct phases of the same substance. Even though there is sure to be a slight loss of energy due to the impact of the steam and water drops when moving at different velocities, this loss is relatively small compared to the extra energy available to accelerate the weight of vapor alone if the drops of liquid are considered to lag behind. The specific volume of the liquid is so very small relative to that of the vapor that it is almost negligible and consequently *the velocity of the vapor has far greater effect on the weight discharged per unit of time from any given area of nozzle than does the velocity of the water.* This phenomenon of two-phase velocities is believed by the authors to be of much greater importance than supersaturation in the study of the flow of steam of low qualities, such as 80 per cent or so.

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[18] A New Theory of the Steam Turbine, by H. M. Martin, Engineering, Vol. 106, 1918.

[19] Sibley Journal of Engineering, Jan., 1923, p. 2.

[20] Dictionary of Applied Physics, Vol. 1, p. 369.

(h) The **combined effect of supersaturation and two-phase velocities** on the weight of wet steam discharged from a nozzle cannot be given quantitatively in its most general form at the present time. However, sufficient work has been done on the subject to show remarkably close agreement between the measured and the calculated discharge in a number of cases studied by the late Professor Goodenough and Professor Keenan at the General Electric Company's plant. By the kind permission of *Power*, Figs. 265, 266 and 267 are reproduced from the instructive article by Professor Goodenough.[21]

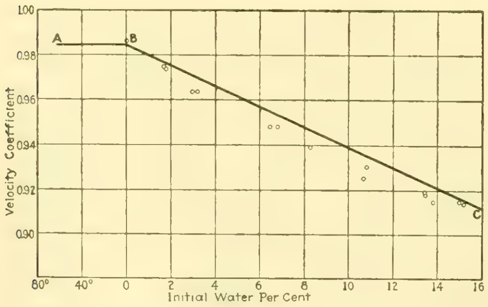

FIG. 265.—Variation of Velocity Coefficient with Initial Moisture Content of Steam.

The points shown in Fig. 265 represent velocity coefficients determined experimentally by measuring the actual jet velocity by means of its reaction and comparing this velocity with that obtained by using

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[21] Supersaturation and the Flow of Wet Steam, by G. A. Goodenough, Power, Oct. 4, 1927, p. 511.

Eq. (318). The solid line *ABC* represents the coefficients for wet steam calculated from Goodenough and Keenan's equation, which takes into consideration the two-phase velocities. The equation is

$$c = K_1\sqrt{x + fy}, \qquad \ldots \ldots \quad (337)$$

in which K_1 = velocity coefficient due to fluid friction between the steam and nozzle;
x = quality of the steam entering the nozzle;
$y = 1 - x$ = fraction of water, initially;
f = (velocity of water drops) ÷ (velocity of vapor).

In their experiments the initial pressure was 31.88 lb. per sq. in. and the discharge pressure was 65 per cent of this. The initial qualities varied over the wide range shown by the figure. For these conditions the value of K_1 was taken as 0.984 and $f = 0.15$. The very remarkable agreement between the experimentally determined points and the calculated coefficients, as given by the line *ABC*, is convincing evidence that Eq. (337) is very close to the truth for this particular case, and probably reasonably accurate in general. The rapid decrease in the velocity coefficient as the percentage of moisture is increased cannot be accounted for by friction between the steam and nozzle, or by supersaturation. On the other hand, Eq. (337) was logically developed by Goodenough and Keenan to account for the action between the steam and water when passing through the nozzle at different velocities, and is, in the opinion of the authors, the most satisfactory explanation of the observed facts that has yet been given.

The *velocity* from an ideal nozzle as already developed (Sect. **252**) assumes that all particles of the fluid across any specified nozzle section have the same velocity and therefore no turbulence; consequently the ideal nozzle permits isentropic expansion of any fluid within it. This conception of the ideal nozzle may continue to be used even for wet steam but, strictly speaking, the velocity coefficient of a nozzle for wet steam is a measure of the combined *imperfections of the nozzle and also of the fluid*, because no matter how nearly perfect a nozzle alone might be any two-phase substance would be unable to have isentropic expansion therein because of the turbulence produced by the unequal velocities of the two phases.

The *weight of wet steam discharged* from a turbine nozzle is almost exactly proportional to the velocity of the *vapor* regardless of the velocity of the liquid, because the volume of the latter is relatively insignificant for nearly all cases. As an illustration, consider a pound of steam with a quality of 90 per cent and a pressure of 20 lb. per sq. in. abs.

The volume of the liquid would be 0.00168 cu. ft. and that of the vapor 18.09 cu. ft., or in other words, the vapor occupies slightly more than 99.9 per cent of the entire volume, even though its weight is only 90 per cent of the total. Clearly then, when the velocity of the vapor is appreciably greater than that of the liquid, the *vapor velocity* must be greater than the *average velocity of the entire fluid, as commonly and easily calculated* for a given amount of available energy. Therefore the actual discharge of wet steam from a nozzle will generally be greater than the calculated value, due to the fact that the *usual method of calculation gives too small a discharge because it fails to consider the two-phase velocities.* The convenience of the common method of calculating the velocity from the ideal nozzle probably justifies its continuation, and the fact that the correct discharge coefficients may be greater than unity need not cause anyone using such coefficients to lose faith in the law of Conservation of Energy or to believe that the actual nozzle is superior to the ideal.

The combined effect of supersaturation and two-phase velocities on the coefficient of discharge of wet steam is shown in Fig. 266 for the same conditions as specified for Fig. 265. The line *ABC* shows the effect of two-phase velocities on the discharge coefficient, c_d, given by the equation,

$$c_d = \frac{k}{\sqrt{x + fy}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (338)$$

where f, x, and y have the same meaning as in Eq. (337), and k represents the constant value of the discharge coefficient that is to be used for any particular nozzle when superheated steam is flowing through it. For the nozzle used in the experiments for which the experimentally determined points are drawn in Fig. 266 the value of k was taken as 0.984. The difference between the line *ABC* and the experimentally determined curve above it represents the effect of supersaturation on the discharge coefficient for this particular case, and it seems fair to consider it a typical one. From this figure it is clear that for dry saturated steam ($y = 0$) entering the nozzle, the coefficient of discharge would be about 1.5 per

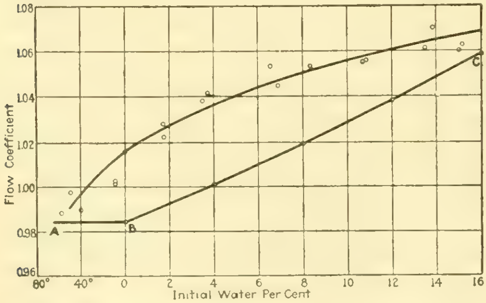

Fig. 266.—The Coefficient of Discharge of a Steam Nozzle as Influenced by Super-saturation and by Moisture in the Steam.

cent greater than unity, and for $y = 12$ per cent, c_d would be 1.06, due chiefly to the two-phase velocities.

(i) **The Energy Losses** in the nozzle through which wet steam is passing would naturally be expected to increase as the percentage of moisture becomes larger because of the greater turbulence produced. Experiments prove that this is true, and the curves in Fig. 267 show the results of Professor Goodenough's calculation for the conditions previously given, all losses being based on the isentropic drop in heat content. The loss due to friction in the nozzle, as shown by curve (a), remains constant at 3.17 per cent for all qualities. The loss due to internal friction, or impact of the steam and water particles, as shown by curve (b), increases directly with the percentage of water. The loss due to supersaturation, as shown by curve (c), remains nearly constant, but decreases somewhat as the moisture becomes greater than 10 per cent. The total energy loss, as shown by curve (d), increases appreciably as the initial moisture becomes greater.

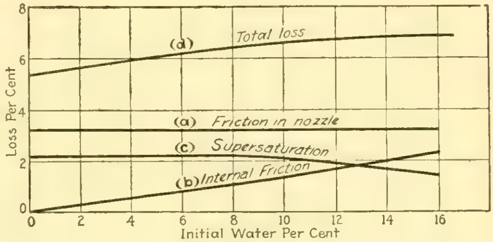

FIG. 267.—Variation in Losses in Steam Expansion with Initial Moisture Content.

255. Nozzle Efficiency.—(a) All real nozzles involve the losses due to fluid friction and turbulence regardless of their cause; but such losses are considered zero in the ideal, which therefore permits isentropic expansion in it. Thus, if the curve ab in Fig. 268 represents the isentropic expansion in the ideal nozzle, the state of the vapor leaving such a nozzle will be represented by b, while b' represents that state at exit of the real nozzle. The less the turbulence that is produced in passing through the actual nozzle, the nearer does the state b' approach b.

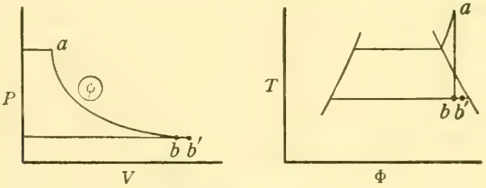

FIG. 268.

Clearly the purpose of a nozzle is to increase the kinetic energy of the jet as much as possible. Consequently the **nozzle efficiency** is defined as

$$e_n = \frac{\text{Increase of K.E. of the jet in actual nozzle}^{22}}{\text{Increase of K.E. of the jet in the corresponding ideal nozzle}}.$$

$$= \frac{\frac{w}{2g}(v_{b'}^2 - v_a^2)}{\frac{w}{2g}(v_b^2 - v_a^2)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (339)$$

But

$$v_{b'}^2 = 50{,}000\,(h_a - h_{b'}) + v_a^2, \quad \text{and} \quad v_b^2 = 50{,}000\,(h_a - h_b) + v_a^2.$$

Therefore,

$$e_n = \frac{h_a - h_{b'}}{h_a - h_b}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (340)$$

(b) The **relation between the nozzle efficiency and the velocity coefficient** is a very simple one, if the entering velocity is negligible. Thus, from Eq. (339), if v_a is zero, then $e_n = v_{b'}^2/v_b^2$, and, from Eq. (335), the velocity coefficient is $c = v_{b'}/v_b$. Hence, for this case the nozzle coefficient is equal to the square root of the nozzle efficiency, or, expressed algebraically,

$$c = \sqrt{e_n} \text{ (if } v_a = 0). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (341)$$

256. Finding the Condition of a Vapor Leaving a Nozzle.—(a) The *heat content of the vapor at exit* from an actual nozzle that is thermally insulated may be expressed in terms of the velocity coefficient, c, the isentropic drop in heat content, and the initial velocity. Thus, referring to Eq. (335) and Fig. 268,

$$v_{b'} = c\sqrt{50{,}000\,(h_a - h_b) + v_a^2},$$

and also

$$v_{b'} = \sqrt{50{,}000\,(h_a - h_{b'}) + v_a^2}.$$

From these two equations, it follows that the *specific heat content* of the vapor at exit from the nozzle is

$$h_{b'} = h_a - c^2(h_a - h_b) + (1 - c^2)v_a^2/50{,}000, \quad . \quad . \quad (342)$$

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[22] In this discussion of the nozzle efficiency the "kinetic energy of the jet" is to be considered *relative* to the nozzle, which may itself have motion opposite to that of the jet. Hence in applying Eq. (339) to moving nozzles the velocities are all relative to the nozzle, which may have a very complex motion relative to the earth. The latter case will be treated more fully in Sect. **258.**

The last term in this equation will always be very small, even when v_a is far above zero, and it may therefore be considered as negligible. Then it follows that

$$h_{b'} = h_a - c^2(h_a - h_b) = h_a - e_n(h_a - h_b). \quad . \quad . \quad (343)$$

With $h_{b'}$ determined from this equation and $p_{b'}$ known, the state b' is completely specified, and the quality, specific volume, and entropy may be found by calculation, or preferably, they may be read directly from a suitable chart, such as Mollier's or Ellenwood's.

(**b**) If the turbulence within a nozzle is very great, the nozzle coefficient becomes very small, and consequently the value of c^2 approaches zero; therefore, from Eq. (343), $h_{b'}$ approaches h_a. In other words, a nozzle becomes a throttling device when it ceases to increase the velocity of the fluid passing through it but, instead, produces disordered motion, or turbulence. On the other hand, the higher the value of the nozzle coefficient, the closer does the expansion in the actual nozzle approach that of the corresponding ideal as represented by the isentropic ab, which is the best expansion line possible in an engine or nozzle that is *thermally insulated*.[23]

257. Forms of Steam Nozzles.—(**a**) Nozzles are made in a considerable variety of forms to suit the space available and to conform to good manufacturing methods. Illustrations of some possible forms of *ideal nozzles* of circular cross-section, designed to permit the flow of 10 lb. of steam per second from an entering pressure of 150 lb. per sq. in. abs. and 100 deg. superheat to a discharge pressure of 6 lb. per sq. in. abs., are given in Fig. 269. For each of the four cases there are also shown the curves giving the velocity, pressure, specific volume, and heat content of the steam during its passage through the nozzle. For each section (weight) × (specific volume) = (area) × (velocity). The velocity at any section is calculated from Eq. (318*c*). Nozzle A has a uniform drop in pressure throughout its length, B has uniform decrease in heat content, C has uniform increase in specific volume, and D has uniform increase in diameter. The jets discharged from the more rapidly flaring nozzles would be unsuitable for use in steam turbines because they have a considerable amount of lateral dispersion. The form of nozzle most commonly used is that shown at D, since it gives a high efficiency and is relatively easy to make. Its divergent portion is conical and approaches somewhat closely the form shown in B, which gives constant acceleration.

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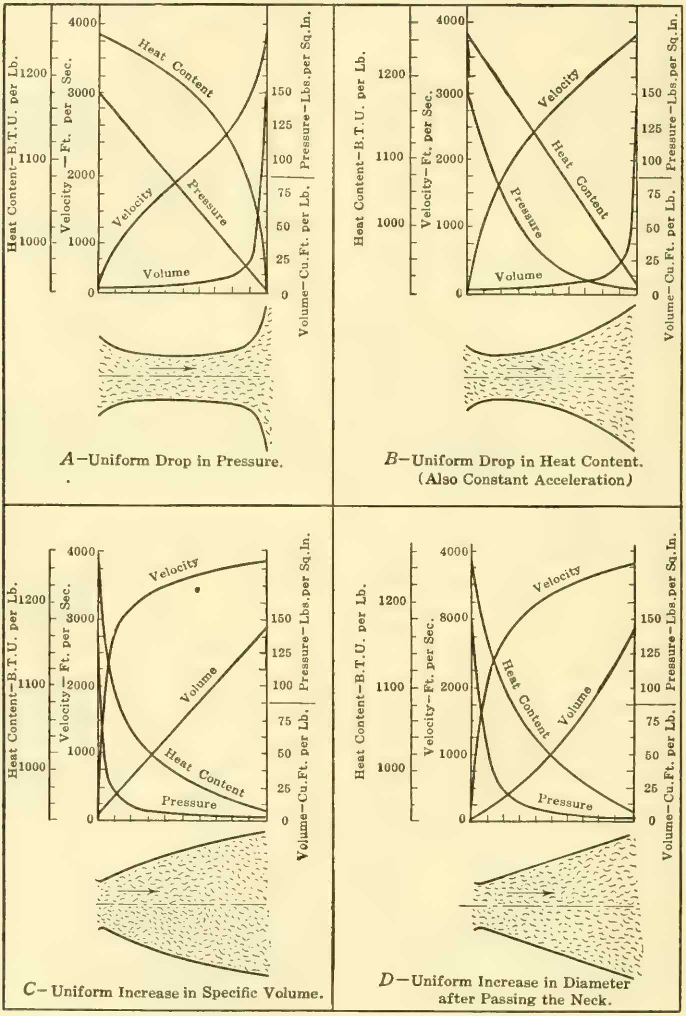
[23] See Sect. **201**, Part I, which is so important that it should now be carefully reviewed by those who are not thoroughly familiar with it.

(**b**) For any discharge pressure higher than that for which these forms were designed, the nozzle may be simply cut off at the point that lies directly below this value of the pressure as shown by the pressure curve.



A—Uniform Drop in Pressure.

B—Uniform Drop in Heat Content. (Also Constant Acceleration)

C—Uniform Increase in Specific Volume.

D—Uniform Increase in Diameter after Passing the Neck.

FIG. 269.

The corresponding values of velocity, specific volume and heat content may also be read directly from their respective curves for any pressure within the range used.

(**c**) **Actual nozzles** used in impulse steam turbines resemble in their simplest form the one shown in Fig. 270, assuming that the expansion is carried to a pressure considerably below the critical. If the nozzle section is circular the diverging part is *conical*, and if this section is rectangular this part is *pyramidal.* In such cases it is merely necessary to compute the neck and end areas of the nozzle and then make the length such as will neither cause the flare to be so great as to prevent proper guidance of the jet, nor make the nozzle needlessly long, thus introducing additional surface against which the jet rubs and requiring greater space for the nozzle than is necessary. Usually the nozzle is oblique with respect to the plane of motion of the turbine blades, and in that case a cylindrical (or rectangular) portion is added, as in the nozzles shown in Fig. 271, to guide the issuing jet and protect its surface stream lines from being diverted by the more-or-less stagnant surrounding vapor. Often it is necessary to curve the axis of the nozzle; and when several nozzles are used they are frequently of rectangular cross-section and are banked closely together, with thin partitions between, so as to form a nozzle block, as shown at (*b*) in the figure. In the diaphragm shown at (*c*) is a set of converging nozzles (with parallel extensions) that consist merely of the passages between guiding vanes of sheet metal that are cast in place. When the volume of steam flowing is large the impulse nozzles may extend around the full circumference of the wheel; and in that case there is said to be a " full peripheral discharge." The nozzles here illustrated apply to impulse turbines, for which many other arrangements are also used. Reaction turbines have both rotating and stationary nozzles, always with full peripheral

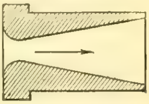


Fig. 270.

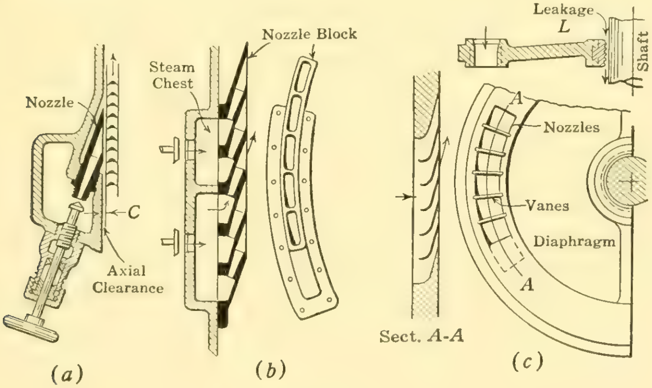


Fig. 271.

discharge. These nozzles are convergent and are formed, as shown in Figs. 273 and 280, in the contracting passages between blades.

Nozzles must be made of *materials* that resist corrosion and erosion; and, when used with superheated steam, the material must also be able to withstand high temperatures. Their passages should be designed and constructed with great accuracy to accord with the particular conditions under which they are to operate. A poorly designed nozzle, or one originally proportioned for other conditions, may cause a very unsatisfactory performance of the turbine. With throttle governing, the nozzles are not operating with their maximum efficiency except at the load for which they are designed; hence, when the operating conditions are suitable, the regulation is often accomplished by "cutting in or out" those nozzles or groups of nozzles that are not needed, this being accomplished by means of valves that are controlled by hand or by governor.[24] Good steam nozzles have efficiencies ranging from 0.85 to nearly unity, depending on the conditions under which they operate. Their discharge velocities range from 400 to 4000 ft. per sec. or more.

258. The Dynamics of Turbine Blades.—(a) After the stationary nozzle has converted some of the heat content of the steam into the kinetic energy of the issuing jet, the latter is made to act on the turbine blades which move at high velocity and which should absorb this jet energy as fully as is possible and then transmit it to the turbine shaft.

In so-called "*impulse*" turbines, the jet, upon coming into contact with the blades, produces on them a force commonly called "impulse,"[25] but since it leaves the blades with a partially reversed motion it also causes a *reaction* on them. In *reaction* turbines, not only are the blades subjected to forces of action and reaction similar to those on impulse blades, but there is an additional reaction on them caused by the acceleration of the jet (produced by the expansion of the steam) while passing rearward over them. Obviously the terms "impulse blading" and "reaction blading" are incomplete, but they have become firmly established by long use.

In studying the dynamic action of the steam jet on blading, it is necessary to analyze the velocities of the blades and the various parts of the jets, both in space and with respect to each other. The movement of any body can be expressed only in terms of its motion relative to some other body. In dealing with the flow of a fluid through the "moving" and "stationary" elements of a turbine, a clear understanding of the terms *absolute* and *relative velocities* becomes necessary. The turbine frame is considered stationary and all velocities referred to it are called **absolute velocities,** even though the turbine may be mounted on a moving steamship or locomotive. When a fluid passes through a moving blade (or bucket or vane) the velocity of this fluid relative to this blade is called its **relative velocity.**

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[24] This, however, on a multi-stage turbine, is usually applied to the first stage only, the following stages being subject to the equivalent of throttle governing.

[25] A preferable, but less customary, term for this is the *force of action*.

(b) The **notation** that will be used in expressing the various terms needed in the study of the dynamics of blading will be as follows:

v_1 = absolute velocity of the jet at entrance to the blade, in ft. per sec.;

v_2 = absolute velocity of the jet at exit from the blade, in ft. per sec.;

R_1(or v_{r_1}) = relative velocity of the jet at entrance to the blade, in ft. per sec.;

R_2(or v_{r_2}) = relative velocity of the jet at exit from the blade, in ft. per sec.;

u(or v_b) = peripheral velocity of the blade, in ft. per sec.;

h_1 = specific heat content of the fluid at entrance to the blade, in B.t.u. per lb.;

h_2 = specific heat content of the fluid at exit from the blade, in B.t.u. per lb.;

w = weight of fluid flowing through the blade, in lb. per sec.

Similar use of the subscripts 1 and 2 may be made with the usual symbols for pressure, temperature and specific volume.[26]

(c) **The increase in kinetic energy of a jet relative to a stationary blade, or nozzle,** that is thermally insulated, may be obtained directly from the general energy equation of steady flow. Thus, this increase in velocity energy, in B.t.u. per sec., is

$$w(v_2^2 - v_1^2)/50{,}000 = w(h_1 - h_2). \quad . \quad . \quad . \quad . \quad (344)$$

Note that $w(h_1 - h_2)$ represents the *actual drop* in the heat content of the fluid in passing through the blade. This energy can equal the isentropic drop only if the flow is without friction or turbulence.

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[26] The selection of suitable symbols for the relative jet velocities and the blade velocity is difficult to make. Strict consistency requires the use of v for all velocities, but this necessitates the use of double subscripts which are troublesome; consequently the letters R and u will be preferred in this text. The small letter r cannot well be used for any velocity because it is often needed to designate the radius of rotation of the bucket, whereas no confusion is likely to arise with the use of R even though it is also used for the gas constant.

If the same amount of fluid should flow through a similar blade passage, or nozzle, mounted upon a rotating disc the *energy transformations between the fluid and the blade* would be just the same as before, provided that the fluid enters and leaves with the same values of h_1 and h_2 as previously, and provided, further, that the entering and exit *velocities of the fluid relative to the moving blade* are the same as the corresponding absolute velocities for the stationary nozzle. The heat content of a jet of any fluid is, by definition, in no way dependent upon its jet velocity. Therefore, *the increase in the kinetic energy of the jet relative to any* **moving** *nozzle* or *blade,* that is thermally insulated, in B.t.u. per second, is

$$w(R_2^2 - R_1^2)/50{,}000 = w(h_1 - h_2). \quad . \quad . \quad . \qquad (345)$$

Various applications of this important relation will be made in the further study of turbine buckets.

(d) **The use of velocity diagrams** is essential to show the relation between the absolute and the relative velocities in a turbine. In such a diagram the velocities are represented both in magnitude and direction by straight lines called *velocity vectors.* For all cases, the *absolute velocity of the fluid is equal to the vector sum of the blade velocity and the relative velocity of the fluid with respect to this blade.* Thus, in Fig. 272, with the blades shaped as shown and with a relative entrance velocity (R_1) having the magnitude and direction indicated, the absolute velocity at the entrance to the blade should correspond to the vector v_1, when the blade has the velocity u.

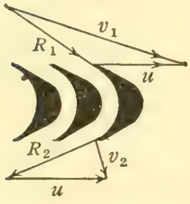


Fig. 272.

With the values of the blade velocity, u, and the relative discharge velocity, R_2, known in direction and amount, the absolute velocity, v_2, of the jet *leaving* the blade is found by constructing the exit velocity triangle shown. With the jet entering the blade in the direction of the vector R_1 there is a minimum of shock, and of energy loss resulting therefrom. Vector R_2 is in line with the direction of the blade surface at exit.

Pure "impulse" blades (Fig. 272), have equal pressures at their entrance and exit edges. If there is no friction or turbulence in the steam passing through the passages between the blades, as with *ideal impulse blades,* then $h_2 = h_1$ and consequently, from Eq. (345), $R_2 = R_1$. *Actual impulse blades,* however, cause R_2 to be from 5 to 15 per cent less than R_1, because of the turbulence and fluid friction occurring in the passages between the blades; therefore h_2 is greater than h_1.

The absolute velocity, v_1, from a stationary nozzle is calculated using the nozzle coefficient and the isentropic drop in heat content in the nozzle for the given conditions. The stationary nozzle performance determines the value of h_1 at entrance to the succeeding moving blade. Then, with the proper values of the nozzle and blade coefficients known, the value of h_2 can be computed by the aid of the vector velocity diagram and Eq. (345).

With *pure "reaction" blades,* p_2 is considerably less than p_1, and consequently h_2 is appreciably less than h_1, even though there may be some turbulence and friction; therefore, from Eq. (345), R_2 is larger than R_1. The vector diagram for this type of blade is illustrated in Fig. 273, but to construct the diagram it is first necessary to determine the values of v_1 and R_2.

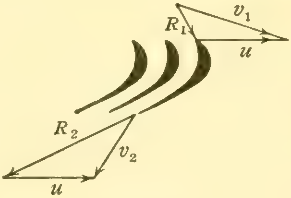


FIG. 273.

The value of v_1 (also h_1) is found from the calculations for the preceding stationary nozzle. Then, with v_1 and u known, the vector triangle at entrance to the blade can be constructed to determine R_1.

With R_1, h_1, p_1, and p_2 known, the value of R_2, and also that of h_2, can be found from Eq. (345) after reducing it to the form,

$$(R_2^2 - R_1^2)/50{,}000 = h_1 - h_2 = (1 - L_f)(\Delta h)_\phi, \quad . \quad (346)$$

in which (Δh_ϕ) is the isentropic drop in heat content from state 1 to p_2, and L_f is the *blade-loss factor* representing the fractional part of $(\Delta h)_\phi$ lost due to fluid friction and turbulence. With L_f known or assumed, R_2 and h_2 can be determined from Eq. (346), and the former can then be used in constructing the vector triangle that enables the direction and magnitude of v_2 to be found.

The ratio $(h_1 - h_2)/(\Delta h)_\phi$, or $1 - L_f$, is sometimes called the "*blade efficiency,*" but since the entering velocity v_1, the blade velocity, and the nozzle and blade angles are all actually involved in the performance of the blade, this term will be reserved for a slightly different use, as shown in paragraph (i) of this Section.

(e) The **total energy delivered to any kind of moving blade** per unit of time will now be considered. This energy is delivered by the blades to the disc or drum on which they are mounted and thus to the turbine shaft from which a small amount of energy is taken to overcome the frictional resistances due to the main bearings, thrust bearings, and also the fluid friction of the rotating parts; but the chief part is delivered by the turbine shaft to the connected machine such as an electric generator. There is no opportunity for much energy to be transmitted by heat or gravity to or from the fluid as it flows through a row of blades;

therefore, by the law of conservation of energy, the actual drop in the heat content of the jet plus its loss in kinetic energy in the time required to pass through a row of moving blades must equal the total energy given to this row in this time.

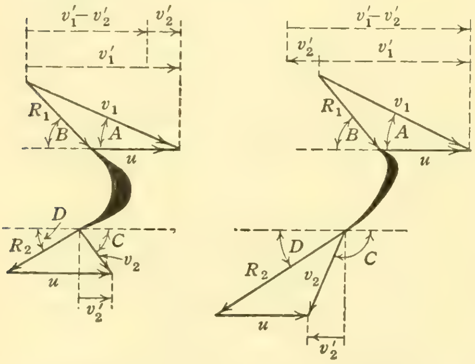
Thus, for a flow of w lb. per sec., the **rate of doing work** upon the blades, in B.t.u. per sec., is

$$Wk_b = w(h_1 - h_2) + w(v_1^2 - v_2^2)/50{,}000. \quad . \quad . \quad . \quad (347)$$

Hence, from Eqs. (345) and (347),

$$\begin{aligned} Wk_b &= [w(R_2^2 - R_1^2) + w(v_1^2 - v_2^2)] \div 50{,}000 \\ &= w[(v_1^2 - v_2^2) - (R_1^2 - R_2^2)] \div 50{,}000. \quad . \quad . \quad (348) \end{aligned}$$

(f) **Another method** of finding the work done upon the blades in a unit of time will now be given. The absolute entrance and exit velocities of the jet may both be resolved into *tangential* and *axial components.* The first component is in the direction of motion of the blade and is also called the *peripheral* one; the other one is parallel to the axis of rotation. The **tangential force** transmitted *by the fluid to the blades* equals the mass of fluid flowing per second multiplied by the resultant algebraic decrease in the tangential components of the absolute velocities of this amount of fluid, the direction of motion of the blades being considered positive. The axial force does not produce any turning moment on the shaft, but really interferes with the movement of it by increasing the pressure on the thrust bearing. However, this axial component is necessary to permit the fluid to enter and leave the buckets satisfactorily.



(*a*) (*b*)

FIG. 274.

The tangential components of v_1 and v_2 are represented by the vectors v'_1 and v'_2, respectively, in Fig. 274, in which (*a*) and (*b*) are respectively, impulse and reaction blades. For both cases, (*a*) and (*b*) in this figure, trigonometry shows that

$$v'_1 = v_1 \cos A = u + R_1 \cos B, \quad . \quad . \quad . \quad . \quad (349a)$$

and

$$v'_2 = v_2 \cos C = u - R_2 \cos D. \quad . \quad . \quad . \quad . \quad (349b)$$

Hence the **tangential force** on the blade when w lb. per sec. are flowing is, in pounds,

$$F_b = \frac{w}{g}(v'_1 - v'_2) = \frac{w}{g}(v_1 \cos A - v_2 \cos C) \quad . \quad . \quad (350a)$$

$$= \frac{w}{g}(R_1 \cos B + R_2 \cos D). \quad . \quad (350b)$$

These equations show that, not only is F_b equal to w/g times the algebraic difference between the tangential components of v_1 and v_2, but it is also equal to the product of the mass and the algebraic difference between the tangential components of R_1 and R_2.

For fixed values of v_1 and A these equations show, with the aid of Fig. 274, that the tangential force on a given blade becomes larger as the blade speed is reduced, reaching its maximum value when the blade is stationary, the value of v_2 then becoming equal to R_2 in magnitude and direction. By changing the nozzle and blade design so as to reduce the nozzle angle A and the exit angle D the tangential force may be increased. These angles cannot be made zero, however, because the other blades would interfere with the flow of steam.

Since the blades are moving at the constant speed of u ft. per sec., the **rate of doing work** on them, in ft-lb. per sec., is

$$Wk_b = F_b u = \frac{w}{g}(v'_1 - v'_2)u, \quad . \quad . \quad . \quad . \quad . \quad (351a)$$

$$= \frac{w}{g}(v_1 \cos A - v_2 \cos C)u, \quad . \quad . \quad . \quad . \quad (351b)$$

$$= \frac{w}{g}(R_1 \cos B + R_2 \cos D)u. \quad . \quad . \quad . \quad . \quad (351c)$$

These three equations will give the same results as Eqs. (347) and (348) when both are reduced to the same units.

(**g**) The **horsepower** delivered to the blades by the assumed rate of flow is obtained by dividing Eq. (351) by 550, or by dividing Eqs. (347) and (348) by 0.707.

(**h**) The **torque** in lb. ft., delivered to the turbine shaft by a jet of w lb. per sec. acting on one row of blades having a radius of rotation of r ft. is $F_b r$, or

$$\text{Torque} = \frac{w}{g}(v_1 \cos A - v_2 \cos C)r \quad . \quad . \quad . \quad . \quad (352a)$$

$$= \frac{w}{g}(R_1 \cos B + R_2 \cos D)r. \quad . \quad . \quad . \quad (352b)$$

From these equations and Fig. 274 it follows that, for any given blade and a fixed value of v_1, the torque becomes larger as the blade speed is decreased, *reaching its maximum when the blade is stationary.*

(i) The **efficiency of a turbine blade** of any kind is the ratio of the useful energy absorbed by the blade to the energy available from the corresponding ideal blade. The two possible sources of energy for any turbine blade are

(1) That due to the isentropic drop in the heat content of the jet while passing through the blading, and
(2) That due to the velocity of the jet entering the blade.

Hence the blade efficiency is

$$e_b = \frac{wk_b, \text{ from Eqs. (347), (348), or from Eq. (351)/778}}{w(\Delta h)_\phi + (wv_1^2)/(2g \times 778)} \quad . \; . \quad (353)$$

The isentropic drop in heat content in this equation is taken for the same drop in pressure as occurs in the actual blade and with the same initial state of the fluid as that entering the blade.

(j) For an **impulse blade,** $(\Delta h)_\phi = 0$, since there is no drop of pressure in it, and consequently the **blade efficiency** of this type of blade becomes:

$$e_{ib} = [(v_1^2 - v_2^2) - (R_1^2 - R_2^2)] \div v_1^2. \quad . \; . \; . \quad (354a)$$

As each of these velocities can be obtained directly from the velocity diagram drawn to scale, this efficiency of an impulse blade is sometimes called the "**diagram efficiency.**" The velocities can, of course, also be determined by computation, which is the more accurate method of the two.

From Eq. (351c) and Eq. (353) it follows that for the *impulse* blade this efficiency is also

$$e_{ib} = 2u(R_1 \cos B + R_2 \cos D)/v_1^2. \quad . \; . \; . \; . \; . \quad (354b)$$

If this *impulse blade* is assumed *frictionless,* so that $R_1 = R_2$, and if the relative angle B is equal to the exit angle D, then Eq. (354b) becomes

$$\begin{aligned} e_{ib} &= (4uR_1 \cos B)/v_1^2 = 4u(v_1 \cos A - u)/v_1^2 \\ &= 4u(v_1 \cos A - u)/v_1^2 = 4\rho \cos A - 4\rho^2. \quad . \; . \quad (354c) \end{aligned}$$

in which $\rho = \dfrac{u}{v_1}$, and is called the **velocity ratio.** The magnitude of this ratio has an important bearing on the efficiency of the turbine, hence the turbine designer gives it careful consideration.

To find the value of the velocity ratio giving maximum efficiency with frictionless *impulse* blades, differentiate this equation with respect to ρ and equate to zero. This gives:

$$0 = 4 \cos A - 8\rho,$$

thus, the ideal velocity ratio with such blades is:

$$\rho_{\text{ideal}} = \left(\frac{u}{v_1}\right)_{\text{ideal}} = \frac{\cos A}{2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (355)$$

in which case the residual velocity, v_2, is at right angles to u and is as short as possible.

The **maximum efficiency** for such an ideal impulse blade therefore becomes, from Eqs. (354*c*) and (355), equal to $\cos^2 A$. This gives a value of 88 per cent for an angle of 20°, which is commonly used. If the angle is reduced to 12°, which is about the limit, this efficiency is 0.957. The energy loss involved in such a blade is inherent, being due to the necessity of making the angle A greater than zero in the actual construction of the turbine.

The relation between the velocity ratio and the diagram efficiency, as given by Eq. (354*c*) for the frictionless impulse blade, is shown by the solid lines in Fig. 275 for the two values of the nozzle angle A indicated. The broken curve shows the efficiency when the *frictional losses* in the blade have been included, the nozzle angle being 20°. These *frictional losses* are chiefly due to displacing stagnant vapor from between the blades, friction between the jet and the blade, eddying, spilling over the tips of the blades, spattering from impact on the working surface of the blade when an improper entrance angle is used, and impact on the back of the blades. These losses cause R_2 and v_2 to be reduced, and the efficiency may then be expressed as $b \cos^2 A$, in which b is a suitable coefficient, with values from 90 to 98 per cent, to correct for the frictional losses. For maximum efficiency the velocity ratio will still have the same value as without frictional losses, but v_2 will not be quite at a right angle with u.[27]

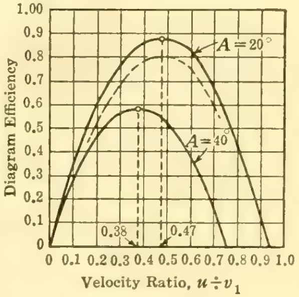


FIG. 275.

(k) For the **reaction blade,** which really has both impulse (action) and reaction in modern turbines of this type, the *blade efficiency* may also be obtained by using Eq. (353), when the velocities and angles are known. However, for any *actual* case the expression for efficiency cannot be reduced to any simple *general* form. The available energy for such a blade, as given by the denominator of Eq. (353), comes from the actual entering velocity v_1 and from the isentropic drop in heat content possible with an ideal blade for the given drop in pressure.

---

[27] When the blade losses are included in the analysis the maximum efficiency also becomes $(1 + R_2/R_1)(\cos^2 A)/2$.

For a *frictionless blade* $(\Delta h)_\phi$ in Eq. (353) becomes

$$(R_2^2 - R_1^2)/2g\,778.$$

Now if it be assumed that this amount of energy is just equal to the kinetic energy of the entering jet, or that $R_2^2 - R_1^2 = v_1^2$, then, from Eq. (353), the efficiency of this type of reaction blading becomes

$$e_{rb} = \frac{(R_1 \cos B + R_2 \cos D)2u}{(R_2^2 - R_1^2) + v_1^2}$$

$$= [(v_1 \cos A - u + R_2 \cos D)2u] \div 2v_1^2. \quad . \quad . \quad (356a)$$

If the velocities and angles are so related that $R_2 \cos D = v_1 \cos A$, then:

$$e_{rb} = (2v_1 \cos A - u)u/v_1^2 = 2\rho \cos A - \rho^2. \quad . \quad . \quad (356b)$$

For this efficiency to be a maximum,

$$\frac{de}{d\rho} = 2 \cos A - 2\rho = 0.$$

Hence the **ideal velocity ratio** with *reaction* blades is

$$\rho_{\text{ideal}} = \left(\frac{u}{v_1}\right)_{\text{ideal}} = \cos A, \quad . \quad . \quad . \quad . \quad . \quad . \quad (357)$$

which causes v_2 to be at right angles to u. This value of ρ is twice that given in Eq. (355) for the impulse blade.

The **maximum efficiency** of the reaction blade becomes, for Eqs. (356*b*) and (357), equal to $\cos^2 A$, which is the same as for the impulse blade.

(1) In **all cases** the efficiency of blades can be increased by reducing the angles A and D and by decreasing the absolute exit velocity v_2. If these angles could be made zero the efficiency of *impulse* blades could be raised to unity by making $u = v_1/2 = R_2$, and that of *reaction* blades would have the same value if $u = v_1 = R_2$, for then, in both cases, the absolute residual velocity v_2 would be zero and all the available energy would be absorbed by the blades, assuming none dissipated

by fluid friction or turbulence. But in the actual turbine, if the angles are made too small, the blades would have to be inordinately long to provide sufficient passage area between them. Hence there are practical limits to the reduction of these angles; and, as a result, the blade efficiency is inherently less than unity. For example, if $A = D = 20°$, then a frictionless *impulse* blade should have entrance velocity $v_1 = u/(\frac{1}{2} \cos A) = u/0.47$, and its efficiency would be 0.88, and a *reaction* blade, for the conditions given with Eq. (357), should have $v_1 = u/\cos A = u/0.94$ and its efficiency would also be 0.88, there being an energy loss of 12 per cent in both cases, merely due to the obliquity of the jets. The actual value of v_1 is commonly made a little larger than the ideal, i.e., $u/v_1 < \rho_{\text{ideal}}$, because a slight increase affects the efficiency but little, as will be seen from Fig. 275, whereas the size and cost of the turbine for a given capacity is reduced considerably thereby. Angles somewhat smaller than 20° are sometimes used, but often larger ones are necessary, especially in the last stages of a multi-stage unit.

(**m**) Obviously, the greater the bucket speed adopted the smaller will be the turbine for a given output, but this velocity must be kept within the limits fixed by the structural strength of the revolving discs or drums, the blades themselves and also the attachment of them to the rotor. A blade speed, u, of 600 ft. per sec. was formerly about the safe limit, but with improved materials and constructions double this value is sometimes used now in large units. Even with such values of u, the drop in heat content per stage, using proper values of ρ, is small, hence if the range of pressure in the turbine is large, the unit must have many stages.

259. Multi-staging.—(**a**) When large drops in the heat content of the steam are to be utilized, it is necessary to multi-stage the turbine because blade speeds are limited by the strength of materials used, and also because suitable turbine shaft speeds are desired to drive, without gearing, various types of machines the speeds of which are much below those of single-stage turbines.

(**b**) **Simple pressure staging** (of the Rateau or Zoelly type) with *impulse* blades is shown in Fig. 276, each stage constituting an elementary impulse turbine with a small pressure drop. As the steam traverses the turbine it expands by increments in the successive nozzles and increases in volume, hence the nozzle areas and blade lengths must

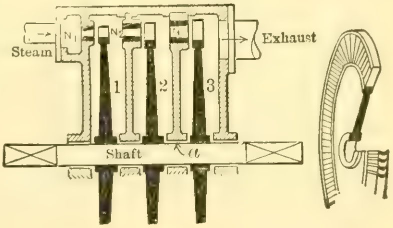


FIG. 276.

increase in like manner throughout the series of stages. By properly proportioning the cross-sectional areas of the inlet and outlet nozzles of any stage the designer can fix at any desired value the pressure to be maintained within that stage.

If there are m pressure stages, then each one will use $1/m$th of the total available energy, assuming an equal distribution of work, and the nozzle discharge velocity and rotative speed would be $1/\sqrt{m}$th of those of a single-stage turbine using the same values of r and ρ. Thus by using nine stages, the turbine speed would be one-third that of the simple unit.

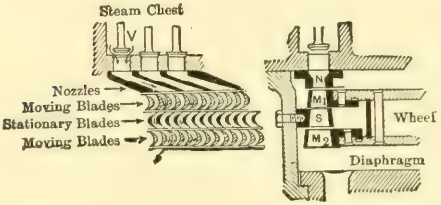


FIG. 277.

(c) **Velocity Staging or Compounding,** patented by Curtis in 1896, is shown in Fig. 277, the exit jet from the first row of moving blades M_1 being redirected by stationary blades S to act on a second row of moving blades M_2 so that the latter will absorb some of the residual energy of the steam leaving the first row. Blade friction and other losses, as well as the cost of construction, make inexpedient the use of more than two or three rows of rotating blades per pressure stage. The velocity diagrams for the stage illustrated in Fig. 277 are given in Fig. 278, the final residual velocity being v_4, which is much less in amount than would result with a single row of blades. In diagrams of this kind $R_1 = R_2$, $R_3 = R_4$ and $v_2 = v_3$ only when there are no losses in passing through the moving and stationary blades. Since the steam is expanded fully in the nozzles, the pressure throughout the casing of the stage is uniform. Compounding of this kind is used extensively in small turbines and in the first stage of large units.

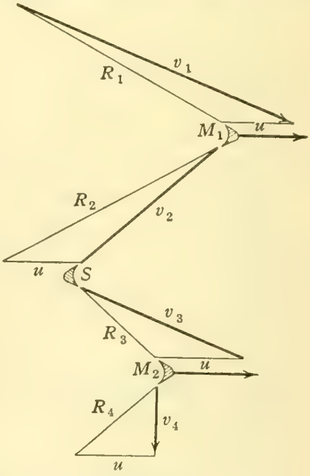
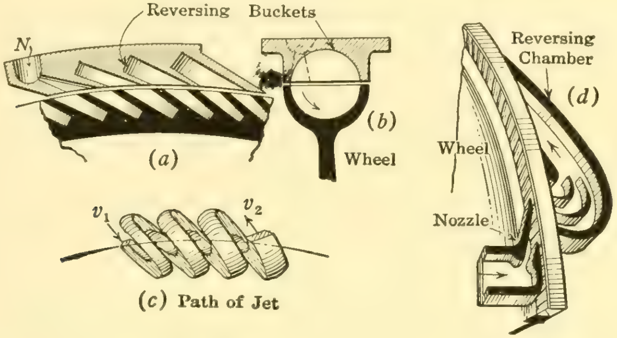


FIG. 278.

If in any pressure stage the velocity is compounded n times with symmetrical and frictionless blades, the maximum blade efficiency is still $\cos^2 A$, but the corresponding velocity ratio becomes $\rho = (\cos A)/2n$, or $1/n$th of that necessary with a simple impulse stage, and the rotative speed is reduced in like proportion.

(d) Re-entry velocity-compounding is illustrated in Fig. 279, the jet from the nozzle, after having acted upon a single row of blades or buckets, being redirected by reversing chambers or buckets so as to act again on the *same* row of blades one or more times, the path of the jet usually being approximately *helicoidal* or *sinusoidal.* Such compounding is often used in small, single-wheel auxiliary turbines in which simplicity and low cost are important considerations.



A. Helicoidal. B. Sinusoidal.

FIG. 279.—Re-entry Velocity-Compounding.

(e) The **reaction turbine** must inherently be one having many stages because, with the usual rotative and blade speeds, the high value of velocity ratio, $\rho = u/v_1$, required in this type of machine makes it necessary to have the pressure drops per stage relatively small. In this turbine the converging passages between the blades constitute the nozzles, both stationary and rotating, as shown in Fig. 280 in which the stationary blades are indicated by S and the moving ones by M. After the first stage, the absolute velocity v_2 is more or less perfectly carried over to the entrance of the next set of stationary blades, where the steam is expanded sufficiently to give the velocity v_1 again before entering the next row of moving blades; and so on for succeeding rows.

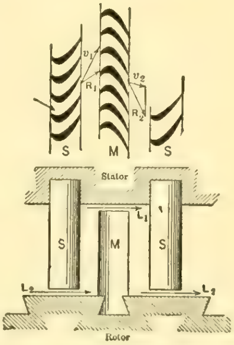


FIG. 280.

(f) **Combinations** of velocity-compounding with pressure-staging in impulse turbines reduce the rotative speeds to $1/(n\sqrt{m})$th of those of simple turbines having the same velocity ratio and diameter of blade wheel. Thus with nine pressure stages, each velocity-compounded twice, the rotative speed could be reduced to one-sixth that of the single stage turbine. Often the velocity-compounding is limited to the first stage, the remaining stages using either simple impulse or reaction blading.

260. Turbine Losses; Reheat and Reheat Factor.—(a) The **energy stream** per pound of steam flowing through a *single stage turbine* is shown in Fig. 281. In this figure the energy supplied per unit weight of steam is $h_a - h_c$, and the available energy, AE, is $h_a - h_b$, where

h_a = the specific heat content of the steam at *admission*;
h_b = the specific heat content of the steam after isentropic expansion to *exhaust* pressure, and
h_c = the specific heat content of the saturated liquid at condenser pressure.

This figure may also be used to represent any single stage of a multi-stage turbine, the terms *admission* and *exhaust* then referring to the *particular stage* under consideration.

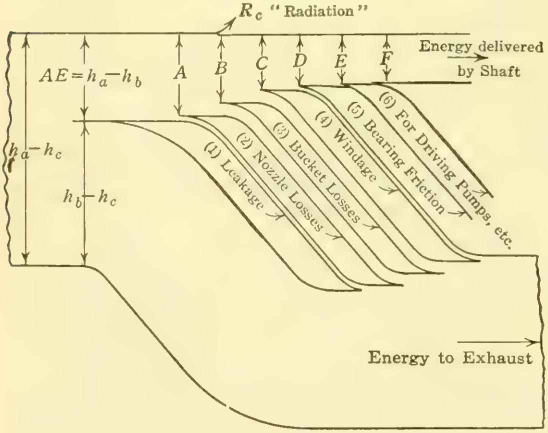


Fig. 281.

(b) The reason why all of the available energy is not delivered by the turbine shaft is due to certain **energy losses** which will now be considered:

(1) **Leakage past the nozzles** may prevent some of the steam from passing through the nozzles. In a multi-stage impulse turbine this leakage may occur through the clearance space between the diaphragm hole and the shaft, and in a reaction turbine it is past the tips of the blades. This loss, which may represent from 0 to 5 per cent of AE, is shown in the figure by stream line (1) and the energy still available for doing work is represented by the diagram width A. There is also a small loss of energy by radiation, conduction and convection marked R_c.

(2) **Frictional resistance and turbulence** in the nozzle will prevent complete conversion of all the energy represented by A into kinetic energy of the jet. These nozzle losses, which are from 2 to 15 per cent

of AE, are represented by stream (2), and the width B shows the kinetic energy produced.

(3) **The blade losses,** consisting of the diagram losses and those due to friction and turbulence in the blades, may waste from 10 to 25 per cent of AE. They are shown by (3) in the figure, thus leaving the width C to répresent the work done on the blades.

In all stages of a multi-stage turbine except the last one, the absolute exit velocity, v_2, of the steam leaving the preceding stage is more or less completely utilized, partially as a "carry over" velocity to the next stage and partially as "reheat" (see (**d**), this section). In the last stage, however, the turbine cannot in any way utilize any of this kinetic energy, which is equal to $v_2^2/50,000$ B.t.u. per lb. This loss is naturally called the **leaving loss,** and is included under the general heading of blade losses because it is one of the diagram losses of the last stage, the blades for which cannot in general be designed to make it zero.

(4) **The windage or rotational losses,** caused by the relative movement of the rotor and its enveloping vapor, prevent some of the energy received by the blades from being delivered to the shaft. Part of this loss is due to churning of the vapor by the *blades* that are not receiving steam from the nozzle. Thus, it is desirable to have the nozzles extend around the full annulus of the blades so that there will be no idle blades to act as fans. This is termed *full peripheral admission.* In *small* turbines this would require using many nozzles that are prohibitively small, and blades which are inconveniently short; hence in such machines the nozzle arc is made only as long as good proportioning permits. In very *large* turbines this limitation wholly or partly disappears. Another portion of the rotational loss is due in part to *surface friction* between *disc surfaces* and the relatively stagnant vapor, and the remainder results from a *radial pumping action* on the film of vapor that is in contact with the disc. The rotational loss is dependent, among other factors, on the vapor density, the square of the diameter of disc, the length and number of rows of blades, and is nearly proportional to the cube of the blade speed. Fortunately, in a condensing turbine the discs having the greatest diameter, longest blades and largest rim speeds are those in the last stages in which the density of the steam is very low. The rotational losses range from 2 to 7 per cent of AE or even more, and are represented in the figure by (4), the energy delivered to the shaft being reduced to the amount D.

(5) The **mechanical losses,** due to friction in the main bearings, and the energy used to drive the oil pumps, governor, and water-seal glands, absorb from 1 to 2 per cent of AE and are represented by (5) and (6), thus leaving the final amount of energy delivered by the shaft as shown

by F. For large turbines, say above 30,000 kw., the mechanical losses are below 1 per cent of the available energy when the turbine is operating at $\frac{3}{4}$ load or more.

The transmission of heat from the turbine casing to the surrounding atmosphere (R_c) is so small that usually it does not need to be considered.

(c) The following efficiencies are often needed when dealing with turbines operating on the Rankine cycle and are stated for convenience in words and in terms of the symbols used in Fig. 281.

The **thermal efficiency** of the turbine is the ratio of the energy delivered by the shaft to that supplied by the steam, or $F \div (h_a - h_c)$. The **engine efficiency** of the turbine is the ratio of the energy delivered by the shaft and the available energy, or $F \div (h_a - h_b)$. The **internal efficiency** of the turbine is the ratio of the energy delivered to the blades and the available energy, or $D \div (h_a - h_b)$. If the total amount of the internal losses, (1), (2), (3), and (4) in Fig. 281, is expressed as i_l per cent of the available energy, the internal efficiency is then equal to $100 - i_l$.

(d) The **reheat** and the **reheat factor** in a multi-stage turbine are important terms used in turbine calculations. All of the internal losses that occur in any stage of a turbine produce turbulence in the steam and thus reduce the work done upon the blades in that stage, but, neglecting the small amount of heat transferred to the turbine casing, these stage losses are absorbed by the steam itself, thus making its quality, volume, entropy, and heat content all *larger than they would be if isentropic expansion had taken place to the same terminal pressure.* The excess of the heat content of the steam, as it actually leaves a stage, over the heat content that would result in this stage from isentropic expansion to the same discharge pressure is called the **reheat** RH, of that stage. Clearly, the reheat is a measure of the turbulence created *within* a stage by its internal losses.

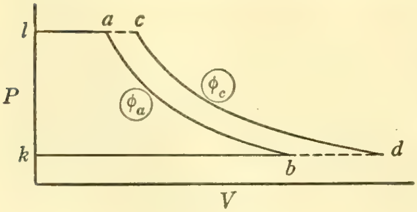


FIG. 282.

The **effect of reheat in any one stage** is to *decrease the energy delivered to the blades of that stage* and to *increase the available energy in the succeeding stage.* Whenever the heat content of a given weight of *any fluid* is increased by any means without changing its pressure, its volume and entropy are also increased. The effect of increasing the heat content of a fluid at a given pressure before it enters any turbine stage in which there is a fixed drop in pressure is to increase the available energy in that stage, as illustrated in Fig. 282. If the fluid enters the stage in state a, in the figure the available energy in this stage is $h_a - h_b$ = area *abkl*, as shown in Sect. **201,** Part I. Similarly, if the fluid enters at the same pressure but with a higher heat content, as at c, the available energy is $h_c - h_d$ = area *cdkl*, which is clearly larger than *abkl* for the same drop in pressure.

This figure also serves to emphasize the fact that the energy available from a fluid in flowing through a turbine is not determined by *pressures alone.*

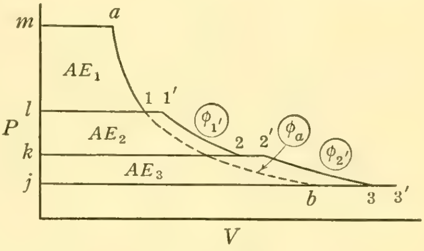


FIG. 283.

(e) Considering the **effect of reheat in a combination of several stages,** refer to Fig. 283, in which the area $abjm = h_a - h_b = AE_{ab}$ = energy available from steam in passing through an ideal turbine regardless of how many stages it may have. Let the corresponding real multi-stage turbine be divided, for example, into three stages in which, due to the reheat, the steam leaves the first stage in state 1′ instead of 1 of the corresponding ideal stage. Because of the reheats in the second and third stages, the steam leaves these stages in states 2′ and 3′, respectively, instead of those shown at 2 and 3. Then, for this multi-stage unit, the *reheats* in the respective stages are:

$$RH_1 = h_{1'} - h_1; \quad RH_2 = h_{2'} - h_2; \quad \text{and} \quad RH_3 = h_{3'} - h_3.$$

The *energies available* are, respectively:

$$AE_1 = h_a - h_1 = \text{area } a1lm$$
$$AE_2 = h_{1'} - h_2 = \text{area } 1'\,2kl$$
$$AE_3 = h_{2'} - h_3 = \text{area } 2'\,3jk$$

. . . .

$$\therefore \quad \Sigma AE = AE_1 + AE_2 + AE_3$$
$$= \text{area } a11'\,22'\,3jm > \text{area } abjm. \qquad (358)$$

The *energies delivered* to the blades are:

$$E_1 = h_a - h_{1'} = AE_1 - RH_1$$
$$E_2 = h_{1'} - h_{2'} = AE_2 - RH_2$$
$$E_3 = h_{2'} - h_{3'} = AE_3 - RH_3$$

. . . .

$$\therefore \quad \Sigma E = E_1 + E_2 + E_3 = h_a - h_{3'}. \qquad (359)$$

The **stage efficiency,** e_s, is the ratio of the energy delivered to the blades to that available in the given stage. It involves the internal losses in the stage. Hence, for a three-stage turbine,

$$e_{s_1} = E_1 \div AE_1 = (h_a - h_{1'}) \div (h_a - h_1)$$
$$e_{s_2} = E_2 \div AE_2 = (h_{1'} - h_{2'}) \div (h_{1'} - h_2)$$
$$e_{s_3} = E_3 \div AE_3 = (h_{2'} - h_{3'}) \div (h_{2'} - h_3)$$

For a single-stage turbine the stage efficiency is the same as its internal efficiency, but this is not true for a multi-stage turbine, as shown by Eq. (362*b*).

(f) **For the entire turbine,** the following relations hold, referring to Fig. 283:

The *reheat* [28] is

$$RH = h_{3'} - h_b = \begin{pmatrix}\text{isentropic}\\ \text{drop in } h\end{pmatrix} - \begin{pmatrix}\text{actual}\\ \text{drop in } h\end{pmatrix}. \quad . \quad . \qquad (360)$$

The *reheat factor* is

$$RF = \Sigma AE \div AE_{ab} = \frac{\text{area } a11'\,22'\,3jm}{\text{area } abjm}. \quad . \quad . \quad . \qquad (361)$$

The *internal efficiency* is

$$e_{int} = \frac{\Sigma E}{AE_{ab}} = \frac{h_a - h_{3'}}{h_a - h_b} = 1 - \frac{RH}{h_a - h_b}. \quad . \quad . \quad . \qquad (362a)$$

Also, in terms of the reheat factor, RF, and equal stage efficiencies, e_s,

$$e_{int} = \frac{\Sigma E}{AE_{ab}} = \frac{e_s \Sigma AE}{AE_{ab}} = e_s RF. \quad . \quad . \quad . \quad . \qquad (362b)$$

From these equations it follows that for a turbine of n stages each one of which has the same stage efficiency, e_s,

$$RF = \frac{AE_1 + AE_2 + \ldots AE_n}{AE_{ab}} = \left(\frac{E_1 + E_2 + \ldots E_n}{e_s}\right) \div AE_{ab}$$
$$= \frac{\text{actual drop in heat content}}{e_s \text{ (isentropic drop in heat content)}}. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (363)$$

Or,

$$\text{actual drop} = e_s(RF)(AE_{ab}). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (364)$$

Also,

$$RH = AE_{ab} - \text{actual drop in } h$$
$$= AE_{ab}[1 - e_s RF] = AE_{ab}[1 - e_{int}]. \quad . \qquad (365)$$

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[28] Note that $RH < (RH_1 + RH_2 + RH_3)$ for the reason given in (**h**) of this section.

With RH known, from test data for cases similar to that which may be under consideration, the total energy available for doing work in the turbine is, from Eq. (361),

$$\Sigma AE = AE_{ab} \times RF, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (366)$$

and the energy delivered to the blades is, from Eq. (362*b*),

$$\Sigma E = e_{int} \times AE_{ab} = e_s \times RF \times AE_{ab}. \quad . \quad . \quad . \qquad (367)$$

In both of these equations RF is more than unity and is increased by making the number of stages larger and by making the stage efficiency less. Note, however, that there is a distinct disadvantage in increasing ΣAE at the expense of reduced stage efficiencies, because the turbine is made for the sole purpose of delivering mechanical energy; hence each stage should deliver to the blades as much of this energy as possible, instead of producing turbulence in the steam and thereby increasing the available energy in the succeeding stages which use it at a disadvantage because of the approach toward the exhaust pressure.

(g) The **relation between the reheat factor and the number of stages** in a turbine will now be considered. The reheat factor of a turbine is basically defined by Eq. (361), or it may be stated in words as the ratio of the sum of the available energies of the individual stages to the energy available in an ideal turbine having the same entering state of the steam and the same exhaust pressure. Clearly, from Fig. 283, the sum of AE_1, AE_2 and AE_3 is larger than the area *abjm* and therefore the reheat factor is always larger than unity. If the number of stages is increased and the stage efficiency remains the same the reheat factor becomes larger, because the greater the number of stages the sooner does the reheat of one stage become available in the succeeding stages to do useful work before the exhaust pressure is reached. A turbine has, of course, no opportunity to use any of the reheat of the last stage.

The energy delivered to the blades of a three-stage turbine, each stage of which has the same efficiency (e_s), is $e_s(AE_1 + AE_2 + AE_3)$; whereas for the corresponding single-stage turbine having the efficiency e'_s, it is $e'_s(h_a - h_b)$. Consequently, for n stages of equal efficiency,

$$\frac{\Sigma E \text{ (multi-stage)}}{E_b \text{ (single-stage)}} = \frac{e_s(AE_1 + AE_2 + AE_3 + \,.\,.\, AE_n)}{e'_s(h_a - h_b)} = \frac{e_s}{e'_s}\,(RF). \qquad (368)$$

The ratio, $\Sigma E/E_b$, will always be greater than unity even if $e_s = e'_s$, because the reheat factor is larger than unity; but generally $e_s > e'_s$, and consequently the multi-stage turbine has an appreciably higher internal efficiency than has the corresponding single-stage machine. However, the multi-stage turbine, even with an infinite number of stages, each one of which has a small amount of turbulence and therefore some reheat, can never equal, although it may approach, the performance of the corresponding ideal which permits no turbulence whatever.

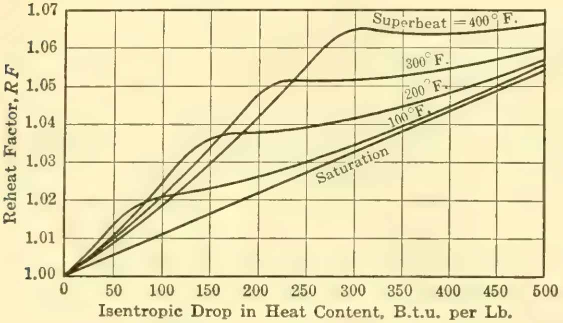

Fig. 284.—Reheat Factors for Steam Turbine with an Infinite Number of Stages. (Closely Approximate for Initial Pressures from 20 to 700 Lb. per Sq. In. Abs.)

The values of the reheat factor for a turbine with an *infinite number* of stages, each one of which has a stage efficiency of *80 per cent,* are given closely by the curves in Fig. 284.[29] If the turbine has n stages, each with a stage efficiency e_s, the reheat factor may be approximately determined by the following equation in which RF_c is the reheat factor read from the curves in Fig. 284:

$$RF - 1 = \left(RF_c - 1\right)\left(1 - \frac{1}{n}\right)\left(\frac{1 - e_s}{0.2}\right) \quad . \quad . \quad . \quad . \quad (369)$$

For example, if $RF_c = 1.05$ this equation gives a reheat factor of 1.067 for a 10-stage turbine with each stage efficiency equal to 70 per cent.

(h) **Steam charts** are almost universally used in making turbine calculations. In Fig. 285, parts (*a*), (*b*), and (*c*), respectively, show portions of the temperature-entropy, Mollier, and Ellenwood charts. In each case the solid line *ab* represents the isentropic expansion of a pound of steam from a known admission state *a* to some given final pressure p_b. If this expansion should be stopped at some higher pressure than p_b, the resultant state of the steam is determined on each chart by the intersection of the new pressure line and the isentropic *ab*. The point b' represents the state of the steam after passing through a stage of a turbine having a stage efficiency of 80 per cent, the reheat being $h_{b'} - h_b$. The point b'' is similar to b' but corresponds to a stage efficiency of 60 per cent, and a reheat of $h_{b''} - h_b$. The intersections of the broken lines ab' and ab'' with the various pressure lines show the respective exit states of the steam that would result if the exhaust pressure from this stage were progressively increased above p_b, the stage efficiencies being kept at 80 and 60 per cent, respectively. Curves similar to ab' or ab'' are often drawn with assumed stage efficiencies for the entire turbine, thus giving the so-called "*condition curves*" of the

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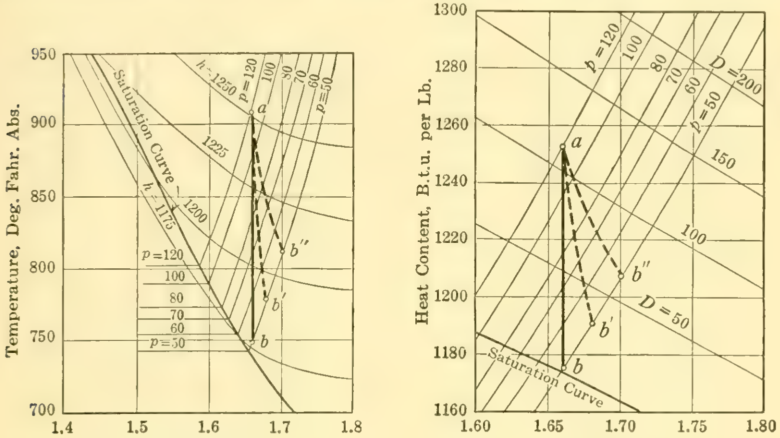
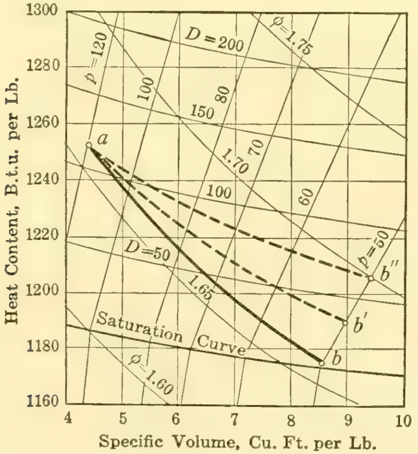
[29] See "Report on Reheat Factors" by Mr. E. L. Robinson, Mech. Eng., Feb., 1928, p. 155. For more accurate values, see "Comparison of Steam Turbine Efficiencies," by the same author, The General Electric Review for July, 1926.



(*a*) $T\phi$ Chart (*b*) Mollier Chart



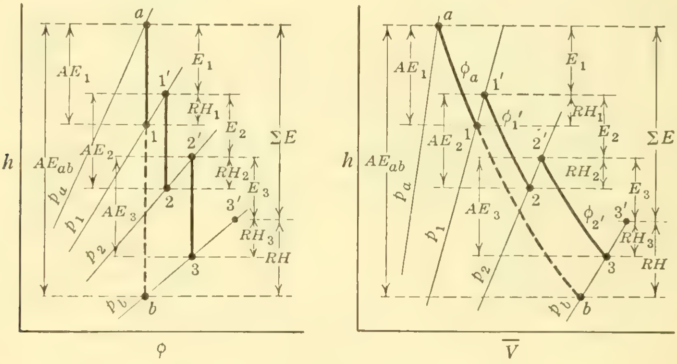
(*c*) Ellenwood Chart

FIG. 285.—Condition Curves.

steam during its passage through the turbine. The actual and isentropic drops in heat content are such important factors in turbine calculations that charts having heat content as one coordinate are more useful in such work than the temperature-entropy chart.

Fig. 286 shows portions of the Mollier and Ellenwood charts applied to the three-stage turbine discussed in a preceding section in connection with Fig. 283. Here $\Sigma E = E_1 + E_2 + E_3 = h_a - h_{3'}$ = the energy delivered to the turbine blades. These charts also show that the *properties of steam cause a divergence of the constant pressure curves as the heat content is increased,* thus explaining why the reheat factor is greater than unity, or why $(h_{1'} - h_2 + h_{2'} - h_3) > (h_1 - h_b)$, which is merely an application of the general principle discussed in connection with Fig. 283. This divergence also shows why

$$(RH_1 + RH_2 + RH_3) > RH, \text{ (or } h_{3'} - h_b).$$



(*a*) Mollier Chart (*b*) Ellenwood Chart

FIG. 286.

(i) The *approximate values of the stage efficiencies* of different kinds of impulse turbines for different velocity ratios as given by Baumann[30] in 1912 are shown by the curves in Fig. 287. These values were obtained from tests on low pressure stages and therefore with relatively low quality steam. With large modern turbines using superheated steam, the stage efficiencies are now appreciably higher than those shown by the curves in this figure; but the chief value of these curves is found in the comparison they afford regarding the different types. Although the efficiency of a Curtis stage may be less than that of the Rateau stage, largely because of the greater blade surfaces and the windage losses involved, it should be remembered that one of the former stages will replace several of the latter and that the curves are therefore not indicative of the relative performances of equivalent combinations.

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[30] Proceedings, I.E.E. (British), 1912, Vol. 48.

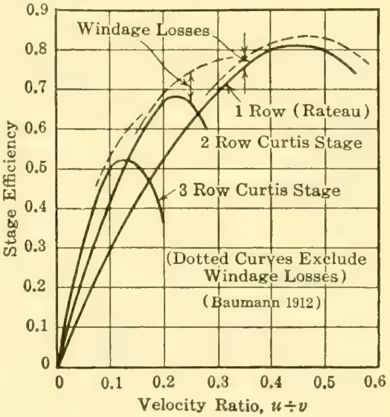

FIG. 287.

Reaction turbines, which are generally built with a larger number of stages than the impulse, have stage efficiencies that lead to substantially the same internal efficiency.[31]

(j) **Electric generators** are very commonly directly connected to steam turbines and the performance of the combination is therefore generally desired. Clearly, if the thermal and engine efficiencies of the turbine, as defined in (c) of this section, are multiplied by the generator efficiency the results will be the *combined thermal* and *engine efficiencies*, respectively.

The approximate values of efficiencies that may be expected from different sizes of turbine-driven generators at various loads are given by Christie[32] as shown in Fig. 288. Additional information regarding the efficiencies of generators of various sizes is given in the following chapter, p. 93.

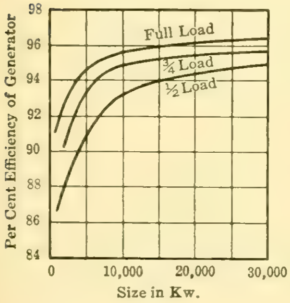

FIG. 288.

261. Turbine Blades.—(a) To prevent spattering and eddying, blades should have their entrance angles equal to the relative angle *B*, in Fig. 274, and must have smooth surfaces of suitable curvature and be so spaced as to form proper guiding passages. When the pressure is reduced, as with governing by throttling, the angle *B* in the velocity diagram changes, hence the entrance angle to the blade should be that best suited to the variable conditions under which the turbine is to operate. To prevent spilling over their tips, impulse blades usually have shroud rings, as in Fig. 252, p. 1. The blade length must be such that the volume of steam flowing between those blades that are opposite nozzles will have the desired velocity, as determined by the velocity diagrams. Because the volumes increase very rapidly when the back pressure approaches the absolute vacuum, the blades in the last rows of a condensing turbine are relatively very long, and with high vacuums the desired length may exceed that which structural strength permits. In that case there must be adopted special expedients which will be considered later; and often some sacrifice of efficiency may be necessary. The design of the blading in the last stages of a large turbine with high vacuum may be the determining feature that fixes the arrangement of the machine as a whole.

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[31] Test data on reaction blading by Stodola are given in Engineering, Oct. 2, 1925, p. 427, and Dec. 18, 1925, p. 775.

[32] Power, Oct. 30, 1923, p. 685.

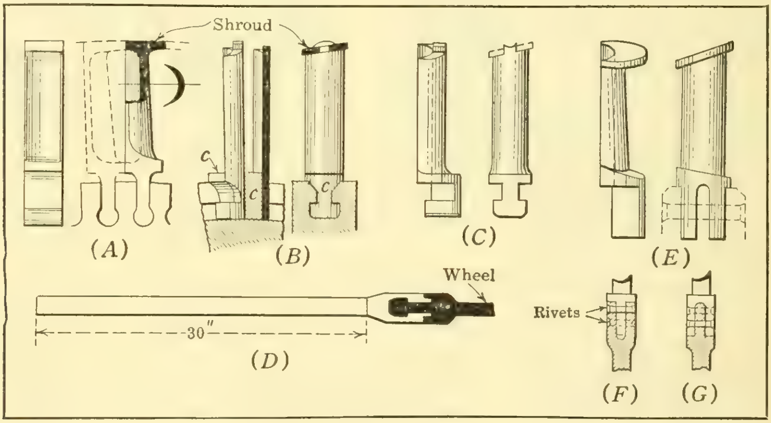

Fig. 289.—Some Impulse Blades.

(**b**) A great many different forms of blade construction have been devised and only a very few can be considered here. The illustrations presented are largely self-explanatory. Fig. 289 gives some examples for *impulse turbines*. The De Laval type of blades shown at (*A*) has a distinctive method of attachment to the disc rim, and the flanged tips form a shroud which adds to the rigidity of construction and prevents the spreading and spilling of the jet. Many other forms of blades are commonly fastened at their roots into dovetail, "T," or similar grooves that are cut in the rims of the discs. Blades that are short and subject to low centrifugal forces are often cut from rolled or extruded bars of the proper cross-section, see (*B*) in the figure, and between them there are inserted soft metal spacers, such as *c*, which also serve as calking pieces. Blades subject to high peripheral speeds are frequently machined from bars, or are drop-forged, in such form as to provide strong thickened bases which may also include the spacers. Some

General Electric blades of this type are shown at (*C*) and (*D*) in this figure. At (*E*) is illustrated a Westinghouse shrouded blade of somewhat similar construction, which in some cases is attached to the rim by means of a T-shank, and in other cases is riveted thereto in the manner shown. Other arrangements of riveting are given at (*F*) and (*G*). Long blades may be tapered to thinner sections at their tips, and should have the metal so distributed as to prevent them from vibrating like tuning forks, as this action may cause serious deterioration of the metal.

(c) A few **blades of the reaction type** are illustrated in Fig. 290. As these blades are thinner than those of the impulse type, long ones are usually inter-stayed by **lashing** or *lacing* wire or bars which are rigidly attached by soldering (often with silver solder), or in other ways. The Westinghouse blades of the form shown at (*A*) are stayed by means of a wire of comma section, the tail of which is depressed between blades. These blades have toes which hook under the calking pieces, *c*, and their tips are "profiled" so as to offer little resistance should rubbing occur. The side wedges *a* and *b* are dispensed with except under severe conditions. Very long blades, such as those shown at (*B*) and (*C*), with round wire lacing, are forged tapered and warped throughout their length so as to vary properly the entrance angles, as *u* has a greater value at the tip than at the base. The Allis-Chalmers blades shown at (*D*) are cast into foundation rings and have shroud rings as well as lacing. The Brown-Boveri blade shown at (*E*) has

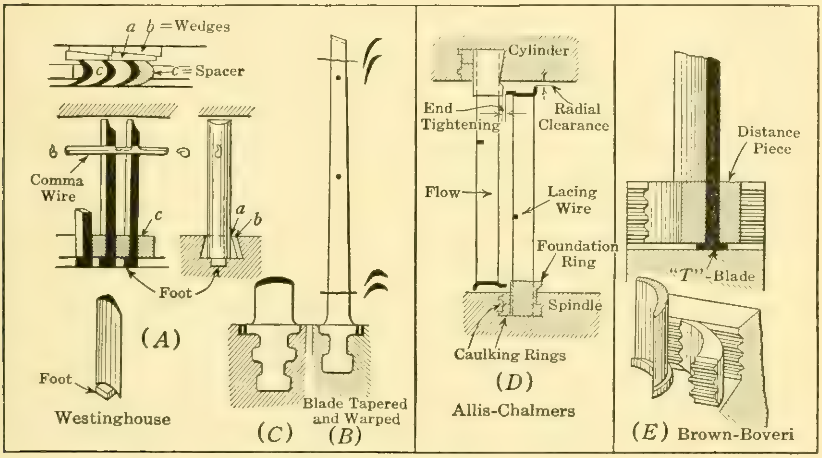

FIG. 290.—Some Reaction Blades.

a T-base which projects under distance pieces which fit into the serrations in the sides of the groove.

The blades in these two figures illustrate [33] only a few of the various types used by the firms mentioned, and blades similar to some of them are also used by other manufacturers. When developing a new turbine the designer selects for each stage that form of blade that appears to him best suited to meet the conditions imposed. A great deal of effort is still being devoted to discovering new and better kinds of blade construction.

(d) The **materials** used for blades should be such as will resist erosion and corrosion, and those subject to superheated steam must be able to withstand very rapid repetitions or constant applications of high temperatures. With saturated steam various copper alloys are often used. For superheated steam copper-nickel combinations, Monel metal, and steels alloyed with large percentages of nickel, chromium, or other noncorrosive materials are commonly employed.

(e) The amount of **blade clearance** used is important, as it affects the turbine performance. In *impulse turbines* the radial clearance at the tips of the blades may be large, but the *axial clearance*, see C in Fig. 271 (a), should be as small as possible and yet allow for inaccuracy of adjustment and for possible warping and vibration of the blades and of the rim of the disc. In *reaction turbines*, to minimize the leakage past the blade tips, see L_1 and L_2 in Fig. 280, the *radial clearance* is reduced as much as structural and operating conditions permit, the axial clearance, however, being relatively large. In (D) of Fig. 290 is shown an arrangement of reaction blades with shroud rings having " end tightening " in the axial direction, in addition to small radial clearance.

Although there is no leakage at the blades and nozzles of the impulse turbine, there is loss of steam through the clearance between the shaft and the diaphragm, see L in Fig. 271 (c), which, with the large pressure drops existing between stages, may approach or equal the blade leakage loss in the reaction turbine.

262. Operating and Critical Speeds, Vibrations and Balancing.—(**a**) Steam turbines are inherently machines that must operate at high rotative speeds to obtain their best thermal performances, and the smaller the unit the higher will this speed be. If used for direct-driving a two-pole alternating current generator having a frequency of 60 cycles per second, there will have to be 60 revolutions per second, or the **operating speed** would be 3600 r.p.m.; whereas the speeds with four- and six-pole machines would be, respectively, 1800 and 1200 r.p.m. Since, for a given output, the higher the r.p.m. used, the lower will be the cost of the turbine and the machine it drives, the rotative speed selected is usually made as great as the structural strength and other considerations permit. Low-speed machines may be driven by high-speed turbines through mechanical reducing gears; or often such machines may be operated by electric motors which receive current from direct-driven generators.

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[33] Some of these blades, like many other details that are described in this chapter, are covered by patents.

(**b**) As **vibrations** due to lack of balance of revolving parts may be serious at the high rotative speeds at which turbines operate, the proper balancing of those parts is imperative. If a shaft having two equal weights that have equal and opposite offsets from its axis is placed on horizontal knife-edges it will be found to be in **static balance**; but should one weight be heavier or have greater leverage than the other, weights must be added or subtracted to effect a balance. But this statically balanced rotor, when revolving at high speed, is not in **dynamic balance,** unless the offset weights are in the same transverse plane; for if the latter is not the case there will be a *centrifugal couple* which may cause very serious disturbances, to overcome which an equal and opposite couple must be introduced by adding or removing other weights. Dynamic balancing may be done by cut-and-try methods, or by the use of special balancing machines. In some cases, a part that is balanced while cold may be out of balance when hot or when its internal strains have been readjusted by repeated heating and stressing in service.

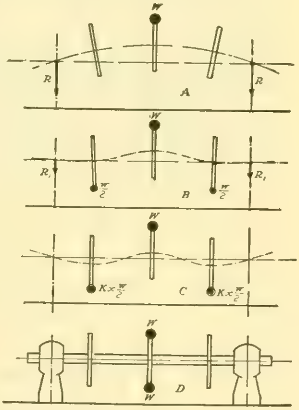

Fig. 291.—Effect of Shaft Deflection Produced by Internal Couples at Speed.

The result of shaft deflection, including the effect on bearing pressures, produced by *internal couples* formed at operating speed is illustrated in Fig. 291, reproduced from the paper [34] by Thomas C. Rathbone.

(**c**) Lack of balance of the turbine rotor not only causes vibration of the turbine itself, but also in the turbine supports, foundations, building, etc. The combination of all parts affected has a natural periodicity which should be made different from the r.p.m. of the turbine or its harmonics. Something of the complications involved when one attempts to reduce the dynamic characteristics of a turbine-generator installation to a simple representative system is shown in Fig. 292.[35] The method of attachment of the steam main and the condenser to the turbine, the kind of foundation, the distance between bearings, the stiffness of the rotating elements between bearings, the alignment of the bearings, and distortions of all these parts due to variation in temperature, combine to make a problem most difficult to analyze. Nevertheless the methods of balancing and the special machines used for that purpose have reached a high degree of refinement.

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[34] Turbine Vibration and Balancing, Trans. A.S.M.E., 1929, A.P.M.-51-23, p. 267.

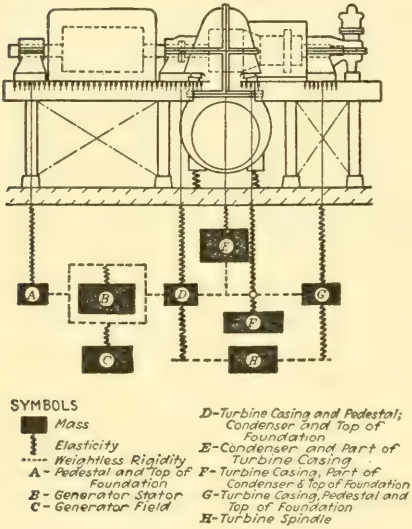
A
B
C
D
E
F
G
H
SYMBOLS
Mass
Elasticity
Weightless Rigidity
A - Pedestal and Top of Foundation
B - Generator Stator
C - Generator Field
D-Turbine Casing and Pedestal; Condenser and Top of Foundation
E-Condenser and Part of Turbine Casing
F- Turbine Casing, Part of Condenser & Top of Foundation
G-Turbine Casing, Pedestal and Top of Foundation
H-Turbine Spindle

Fig. 292.—Diagram of Simplified Vibratory System Representing Turbine-Generator and Supporting System.

(d) Although the rotating parts of a turbine may have been balanced with great care, the gravity axis of a rotor will never exactly coincide with the geometric axis. At low speeds the rotation is about the geometric axis; and at infinite speed it would be about the gravity axis, the shaft then being bowed an amount equal to the original eccentricity of the center of gravity.

If the speed of the rotor is increased gradually from rest, it will eventually reach a value at which the rotation ceases to be about the geometric axis and tends to be about the gravity axis. At a certain speed, called the **critical or resonant speed,** this shifting of axis is usually accompanied by vibrations which may be serious, possibly sufficient to break the shaft, if the speed remains at that value for an appreciable length of time. Resonant speed should be passed through quickly, if at all. Turbines should be operated at speeds which are either considerably above or below the resonant values, or their harmonics. The rotor of a new turbine which is to operate at a given speed should be so designed that its dimensions and distribution of its mass will result in a speed that will not synchronize with the r.p.m. Briefly, the resonant speed coincides with the frequency of oscillation of the rotor when considered as a vibrating beam loaded by its own masses; it is a function of the stiffness or deflection of the beam when loaded by the unbalanced centrifugal forces resulting from the deflection.

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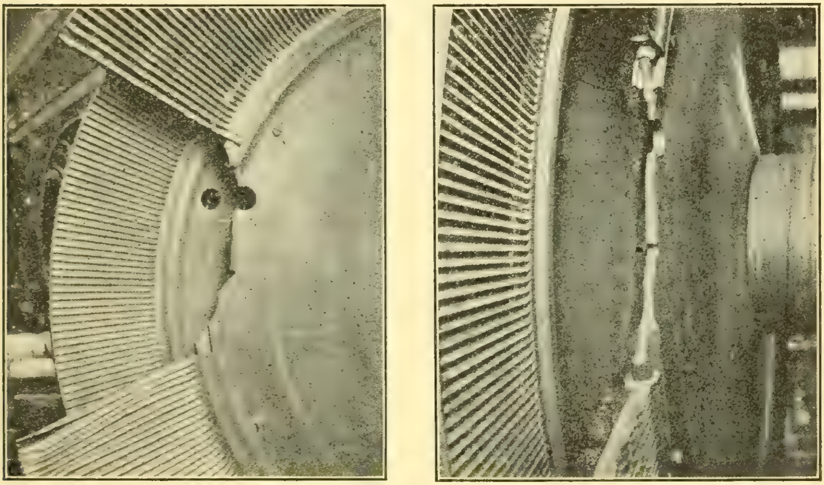
[35] Ibid.

(e) The **vibration of discs and blades** may be serious if these parts are not properly designed and constructed. The rims of large, rapidly revolving and very thin discs may vibrate, or flutter somewhat like a flag in a breeze, and this vibration may possibly cause the eventual disruption of the disc. Also the blades may vibrate like tuning forks and cause deterioration of their metal or fastenings. By the proper distribution of the metal the tendency of these parts to vibrate can be minimized and the frequency of oscillation can be made such as to avoid synchronizing with the rotative speed or with its harmonics. Examples of failure due to vibration and overspeeding are shown in Fig. 293, reproduced from the paper of Wilfred Campbell.[36]



(*a*) (*b*)

FIG. 293.—(*a*) Broken Turbine Bucket Wheel in 9th Stage of 15,000 kw., 1,800 .p.m., 9-Stage Turbine. This Was Caused by Axial Vibration of a Type Which Has Been Eliminated by the Investigation Conducted by Campbell. (*b*) Detail of Fracture in (*a*), Showing Characteristic Fatigue Failure.

(f) A full discussion of balancing and methods of designing for strength and avoidance of critical vibrations of rotor, discs and blades, is beyond the province of this book. These subjects are extremely important to designers and are treated elaborately in the larger texts on steam turbines, in transactions of engineering societies, and in periodical literature. The following additional references are suggested to those needing further information:

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[36] Protection of Steam Turbine Disc Wheels from Axial Vibration, Trans. A.S.M.E., 1924, Vol. 46.

CAMPBELL and HECKMAN, Tangential Vibration of Steam Turbine Buckets. Trans. A.S.M.E., 1925, Vol. 47.
DEN HARTOG, The Mechanics of Plate Rotors for Turbo Generators. Trans. A.S.M.E., 1929, APM–51–1, p. 1.
FRITH and BUCKINGHAM, Vibration in Engineering, 1924. Macdonald and Evans, Publ.
KIMBALL, Vibration Damping, including the Case of Solid Friction. Trans. A.S.M.E., 1929, APM–51–21, p. 227.
KIMBALL and HULL, Vibration Phenomena of a Loaded Unbalanced Shaft While Passing through Its Critical Speed. Trans. A.S.M.E., 1925, Vol. 47.
TIMOSHENKO, Vibration Problems in Engineering, 1928. D. Van Nostrand Co., Publ.

263. Small Turbines.—(a) The **De Laval**[37] turbine was the first of the impulse type to be a commercial success and its development (about 1888) brought out and solved many of the problems encountered later in the design of other turbines. The arrangement of its nozzles and disc is shown in Fig. 252 and the velocity diagram resembles that in Fig. 272. As the velocity (v_1) of the jet issuing from the nozzle may be from 3000 to 4000 ft. per sec., a blade speed (u) which corresponds to maximum efficiency is not used, for no available materials or possible constructions will withstand the resulting stresses. The blade velocities are therefore made as high as is safe. The wheels of the 300 hp. De Laval turbine are about 30 in. in diameter and rotate at about 10,600 r.p.m., with peripheral speed of about 1380 ft. per sec. The 5-hp. turbine has a wheel about 4 in. in diameter, the r.p.m. are 30,000, and the rim speed is 515 ft. per sec.

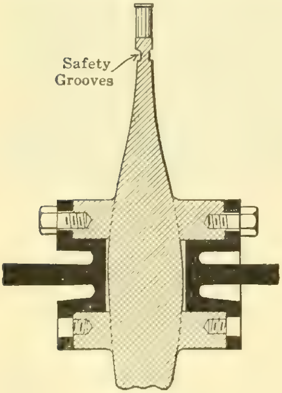

FIG. 294.—Solid Disc.

In the large machines the disc is of the *solid* form shown in Fig. 294, the shaft being made in two parts, each fastened to the side of the wheel by flanges, the centering tongue being a taper fit. The disc is of alloy steel and is designed to have uniform radial and tangential strength. The disc is not pierced for the shaft, because a central hole (or flaw), even if small, will reduce its strength to half. In the small De Laval turbine, such as that illustrated diagrammatically in Fig. 295, the shaft passes through the disc, which has a very large hub to compensate in part for the hole. The disc is also provided with safety grooves near the rim so that in case of accidental overspeeding of serious magnitude (the stress varying as the square of the speed) only a portion of the rim will break; then, because of the resulting centrifugal unbalance, the hubs of the disc will rub against the surrounding portions of the casing and act as brakes to reduce the speed. The flanged blades and their method of attachment are shown at *b*. The shaft is of very small diameter and the bearings *B*, *B* are far apart; thus the rotor tends to gyrate about its gravity axis, the rotative speed being many times the critical value. Owing to the high speed, the torque on the shaft is small, hence the reduction in diameter is permissible.

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[37] The name De Laval is now also attached to other forms of turbines as this company now makes machines of several different designs. In general, most large manufacturers of turbines make several types, each suited to special conditions, the primary patents on the main principles having expired.

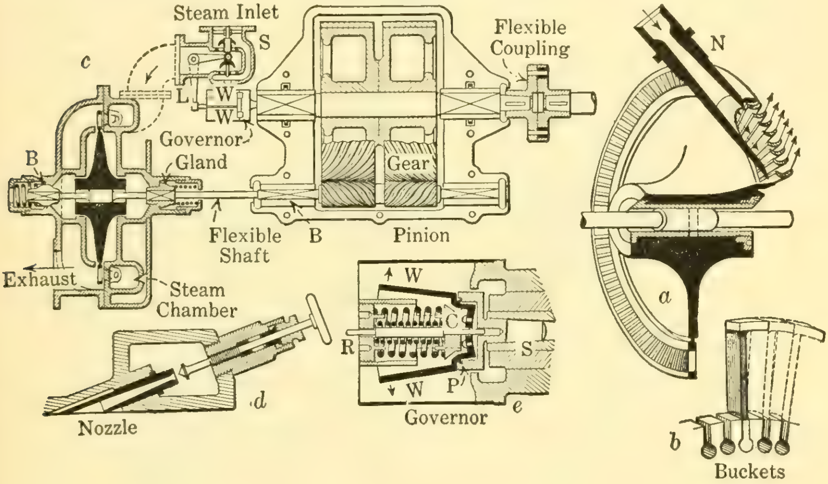

FIG. 295.

In most instances the rotative speeds of these turbines are too great to permit a "direct connection" to the generator, pump, or other machine which is to be driven, hence reducing gears of ratio about 10 : 1 are used. To obtain continuity of action and noiselessness, the gears are of the opposed "herring-bone" type, with very small teeth, which are cut and adjusted with extreme accuracy. The pinion may drive either one or two pairs of large gears, each transmitting power independently. The power is delivered from the gear shaft through a flexible coupling, the bushings shown black in the figure being made of rubber.

The governor, shown at *e*, is of the spring-balanced centrifugal fly-ball type. When the weights *W*, *W* (pivoting on knife-edges at *P*) are rotated rapidly they fly out and the rod *R* is moved longitudinally, thus moving the bell crank *L* (in view (*c*)) and regulating the amount of opening of the governor valve S (which is vertical on actual turbines). Thus the turbine is throttle-governed. Generally there are several nozzles like d around the periphery of the wheel, and these are provided with hand-shut-off valves. If the load on the turbine is small, some of these valves should be closed so that the nozzles remaining in action may operate at or near their maximum capacity (the most efficient condition) instead of having all the valves open with steam greatly throttled. Sometimes there are two sets of nozzles, one to be used when operating condensing, and the other when noncondensing.

(**b**) Other kinds of small turbines have short, stiff shafts and operate *below* the critical speeds, usually at about 3600 r.p.m. A typical rotor for such a machine using Curtis velocity-compounding is illustrated in Fig. 296, with all parts labeled. The main governor is usually mounted, as shown, on the end of the shaft, whereas on larger machines it is on a separate spindle. The emergency, over-speed, or safety governor usually consists of a spring, or spring-balanced weight, which is so adjusted that the centrifugal action will cause it to move outward, say at a speed 10 per cent above normal, and disengage a latch which permits the emergency interceptive valve to close before damage is done by excessive over-speeding. These small turbines have one or more pressure stages, usually with velocity-compounding in the first, and often in all stages, the arrangement of discs and diaphragms of a multi-stage machine being substantially that in Fig. 253, p. 3. Single-stage turbines generally use Curtis, sinusoidal, or helical velocity-staging. With the former, all rows of rotating blades may be on the flange of the same disc, or each row may have its own disc. A typical arrangement of a small single-stage turbine is shown in Fig. 297, which is largely self-explanatory. On such small machines the bearings are self-aligning and ring-oiled, and are located away from the heat of the casing. Often

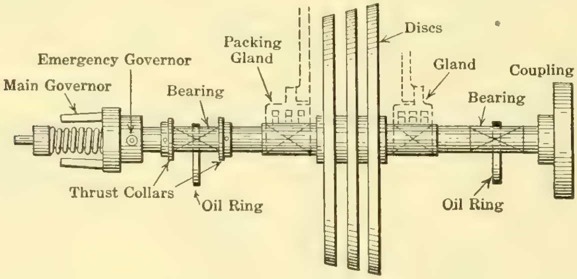

Fig. 296.—Typical Rotor of a Small Turbine with Three-Stage Velocity-Compounding.

all piping connections and all other attachments are made below the center line of the shaft so that the upper half of the casing can be removed without disturbing anything else. When extremely accurate alignment with the driven machine is required, the casing may be supported at points on a level with the shaft, the supports used, such as that at S, being separate from the casing so as not to be affected by temperature changes in the latter. Also three-point support may be used, and sometimes the turbine is rigidly fastened only at one end, the other being flexibly mounted (but properly guided) to allow for changes in length with temperature variations. In addition to the parts shown, there should be a main throttle (or shut-off) valve; and also a strainer should be placed in the supply pipe, to prevent scale or other extraneous solid matter from entering the turbine and possibly injuring the nozzles and blades.

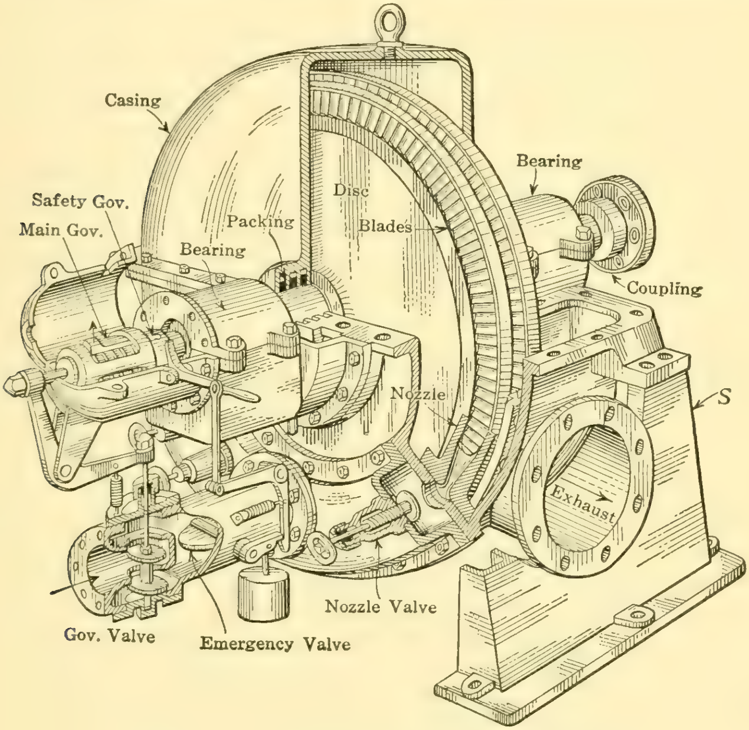

Fig. 297.—Typical Arrangement of Small Turbine.

(c) Small turbines are used for a great variety of purposes: They may be direct-connected, usually through flexible couplings, to centrifugal fans, centrifugal pumps, electric generators, and other high-speed machines. When slow-speed apparatus is to be driven, some form of reducing gear, commonly resembling that of De Laval, is interposed. As there is no internal lubrication, the steam exhausted from the turbine is free from oil and hence is suitable without special treatment for use for various purposes where the presence of oil would be detrimental. Small turbines of the usual commercial type have relatively low thermal efficiencies, but when the heat in the exhaust steam may be usefully employed the economy of the turbine alone is not important. Better turbine efficiencies could be obtained by using more stages and better construction but at increased cost, which is not ordinarily considered expedient.

264. General Features of Medium and Large Single-Cylinder Turbines.—(a) Each turbine manufacturer uses that type of primary element, or that combination of elements, which he believes is best suited to produce a machine that will meet the general requirements with the greatest economy in construction, energy consumption, operation and space, and with the minimum cost feasible; and further, he may modify his standardized forms, or adopt other combinations and arrangements, when special conditions are to be met. Although a great variety of designs are used by competing builders, the final results attained by all, under the same conditions, often differ but little. Only a few designs can be considered here, and it will be assumed in this section that the turbines operate condensing, without bleeding or reheating.

Medium-size turbines may use the same kind of element in all stages, but quite often they are of the combination type that has a Curtis first stage followed either by Rateau or reaction elements in the other stages. In reaction turbines, the high-pressure portion has greater leakage than occurs in the low-pressure stages, because substantially the same radial clearance is used with the shorter blades as with the longer ones, and this portion also requires a relatively long drum; consequently a Curtis element is often substituted for it. This arrangement not only greatly shortens the turbine but also reduces the end thrust so that one or more balance pistons may be omitted.

Large turbines may be made with all stages of the Rateau type, or all with reaction blading, or with a combination of these.

(b) A rather *typical arrangement* of medium-size *impulse* turbine is illustrated in Fig. 253, page 3, with the principal parts labeled. The **rotor** has substantially the same elements as those shown in Fig. 296 for the small turbine, except for the changes due to multi-staging, and except for the removal of the main governor, which in this case is on a separate vertical spindle that is worm-driven.

To allow for a possible offset or obliquity of the shafts of the turbine and driven machine (resulting from faulty erection or the settling of foundations, or due to the curvature of the shafts between bearings as caused by their loading) a **coupling** of considerable flexibility is generally used between the two shafts. That shown (in half) in Fig. 253 is of the type having an intervening sleeve which at each end has pins that fit in bushings of rubber or other suitable material in the holes in the flanges. Other couplings use sleeves, or similar parts, that mesh with jaws, teeth, bars, wires, plates, or similar devices that provide the necessary flexibility. Some shafts which are made in two parts and have only three bearings use rigid flanges which are bolted tightly together, the two outer bearings being raised to allow for the curvature of the two sections of shaft and thus cause the faces of the flanges always to be parallel.

The discs or wheels of an impulse turbine are usually made separate and are forced or shrunk onto the shaft. In a few special cases, however, the shaft and discs are made integral. The spindle or *rotor of a reaction turbine* may consist of a single solid piece, or it may be built up of several parts. In the latter case the rotor may be composed of a cylindrical drum fastened at each end to flanges on the shaft extensions; or there may be a series of discs that are forced on a continuous shaft in such a way as to form the drum.

(c) In place of the multi-collar arrangement of thrust bearing formerly in common use, **thrust bearings** of the Kingsbury[38] type are now often substituted. This bearing consists of a number of stationary segmental plates which are pivoted on hardened steel buttons in such manner that they adjust themselves properly to preserve a wedge of oil-film between them and the single-thrust collar on the shaft, even when under great intensities of bearing pressure. On small reaction turbines, the dummy or balance pistons may be omitted when thrust bearings of this type are used. On impulse turbines, roller or ball-thrust bearings are often used. Thrust bearings are always provided with means for making endwise adjustment of the rotor so that the proper axial clearances may be maintained.

(d) The **governors** on large turbines are not powerful enough to control the steam-regulating valves directly, hence these valves are made to operate through some form of a **relay system,** one arrangement of which is shown in Fig. 298. In this system the control valves of the turbine are moved by a power or operating piston which commonly is actuated by oil under pressure from the oil pump on the turbine, although sometimes steam is used for this purpose. For each load on the turbine there is a definite position of this piston, the position being controlled by the governor acting through a relay or pilot valve on the power cylinder.

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[38] Known in Europe as the Michell thrust bearing.

Governors are commonly provided also with **synchronizing devices** which have auxiliary springs or weights that can be adjusted either directly by hand or remotely by electrical control while the turbine is in operation. These devices make it possible to vary the turbine speed slightly to control the frequency of the generator when bringing the machine into synchronism with other units before connecting it in parallel with them, or for apportioning the load between various units operating together. A general discussion of governors and governing is given in Chapter XXIV.

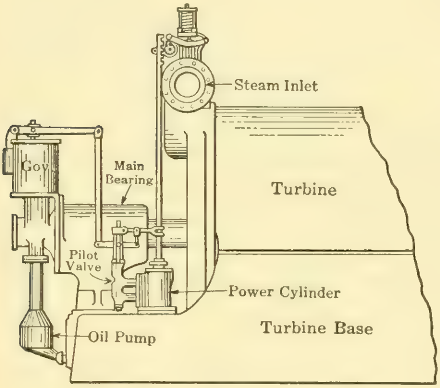


Fig. 298.

When the first stage of the turbine has an impulse element the governing mechanism can be arranged to cut-in or cut-out the nozzles, or groups of nozzles, in such manner that all left in action will function at or near their designed capacity. In Fig. 299 is shown one arrangement whereby loosely mounted valves, each controlling a different group of nozzles, are opened or closed in succession by the movement of the spindle, there being variable amounts of clearance between the shoulders on the valve centers, *A*, *B*, *C*, *D*, etc., and their respective valves. In the arrangement illustrated in Fig. 300 the cams on the cam shaft are set at different angles and each operates a separate nozzle-controlling valve. The number of nozzles in action is determined by the position of the cam shaft, which is regulated by a relay governing system, such as that shown in Fig. 298.

When an impulse turbine is so large that there is normally full peripheral admission, and likewise in all reaction turbines, governor-controlled **secondary valves** are usually provided to meet over-loads. When more power is demanded than can be met by the steam that can pass through the first nozzles or blades, these valves automatically open to one of the lower stages where the passage areas are larger. Sometimes turbines are provided also with tertiary, quaternary, and even quinary valves, which come into operation in succession as the load is increased.

The **safety trip** is not only operated by the emergency governor, but also may be arranged to be automatically disengaged when the pressure of the lubricating oil drops unduly, or when the blade clearance becomes dangerously small. The main throttle valve is usually arranged to close by tripping.

(e) The parts of the **casing**, or cylinder, enclosing the first stages, are subject to high internal pressures and high temperatures, whereas the exhaust-end has low temperatures and a pressure that is *external,* being due to the difference between the atmospheric pressure and the vacuum. The steam chest and those parts of the casing that are subjected to superheated steam are usually steel castings, as cast iron gradually grows in size when repeatedly heated to high temperatures. In recent practice the casing is sometimes built up by welding together steel plates that have been formed to the proper shapes. As the blade clearances are small, the thin and somewhat flexible casing should be designed to provide properly against variations in its shape or length due to pressure, temperature, or load changes. Usually the shell is rigidly attached to the foundation at the exhaust end, and is free to move, with proper guidance, at the other end. It should have adequate drainage. The exhaust chamber is necessarily relatively very large and must be properly ribbed, and perhaps be stayed by rods or plates, so as not to collapse

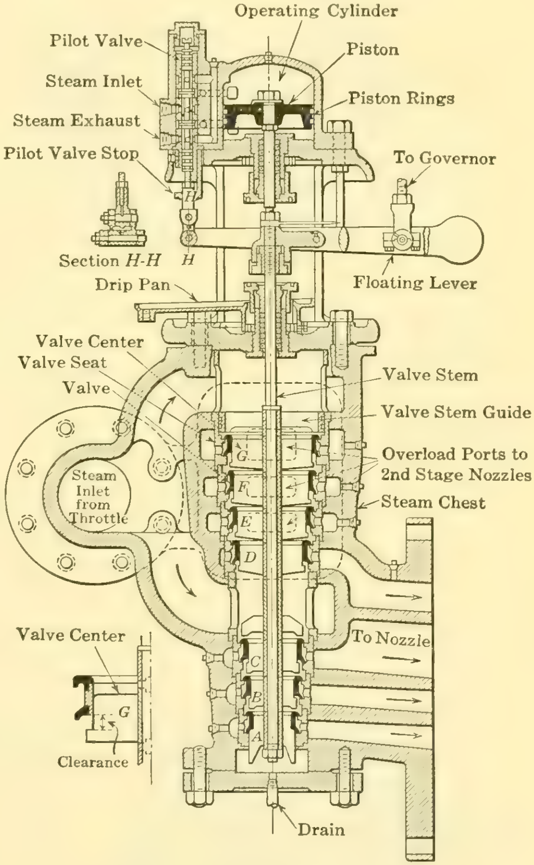


FIG. 299.—One Arrangement of G. E. Steam-Operated Valve Gear.

under the external pressure. The shell should have a sentinel or relief valve to prevent damage if the condenser should fail. The methods of attaching and supporting the steam piping and the condenser, and the connections leading to the latter, should be such as to avoid any possibility of distorting the turbine casing at any time.

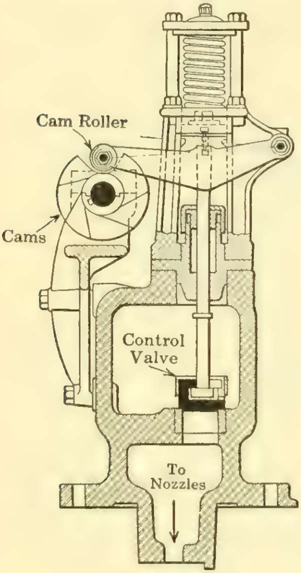


FIG. 300.

(f) There are many forms of **packing** or **sealing glands** used to seal the joints where the shaft passes through the turbine casing. That shown in Fig. 253, p. 3, consists of *metallic or carbon rings* which are composed of segments that are carefully fitted in the grooves of the retaining cages. The rings are prevented from rotating, and the segments are held together in proper position by means of encircling garter springs or by leaf springs on their backs. A few illustrative forms of the *labyrinth type* of packing are shown in Figs. 301 and 302. By means of numerous rings or collars such packing provides a large number of restricted passages to resist the leakage. The *low-pressure* glands of both the carbon ring and labyrinth types always have intermediate grooves that are supplied with steam under a pressure somewhat above atmospheric so that no air can possibly get into the casing and thereby reduce the vacuum. Such steam as does flow through the passages in the *high-pressure* gland is caught in annular grooves near the end of the gland and is piped away for use in the lower stages of the turbine, for heating the feedwater, for sealing the low-pressure gland, or it may be wasted. The water-sealed gland shown in Fig. 303 is substantially a centrif-

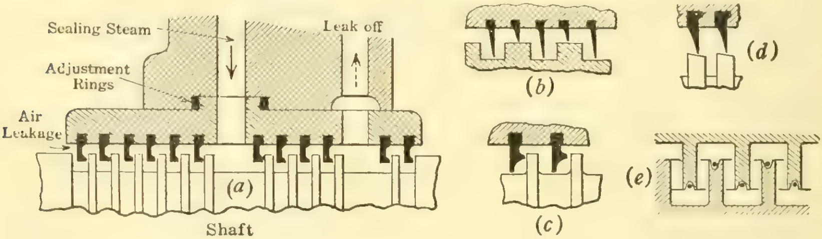
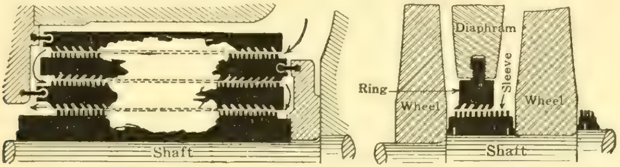


FIG. 301.

ugal pump with a solid impeller which maintains an annular ring of water of sufficient depth and pressure to prevent the passage of air inward. It is sometimes supplemented by a labyrinth packing.

The *diaphragms* between stages of impulse turbines are provided with carbon or metallic rings or with short sections of serrated packing, such as that shown in Fig. 302(*b*). *Balance pistons* in reaction turbines use packing of the labyrinth type.



(*a*) (*b*)

Fig. 302.—One Form of G. E. Labyrinth Packing.

(**g**) On large turbines the **lubrication** is ordinarily by means of a forced feed system in which the oil is repeatedly circulated under pressure by a rotary type of pump which is gear-driven from the turbine or governor shaft. The lubricant is continuously filtered and passed through an oil cooler before being returned to the bearings; also, from time to time, the oil may be passed through centrifuges, or centrifugal separators resembling cream separators. To insure adequate lubrication when starting or when the main oil pump meets with mishap, an additional *independent emergency*, or *auxiliary, oil-pump* is also usually

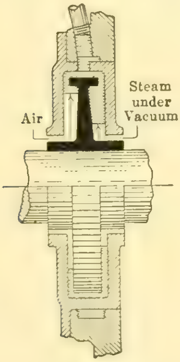


Fig. 303.—Water-Sealed Gland.

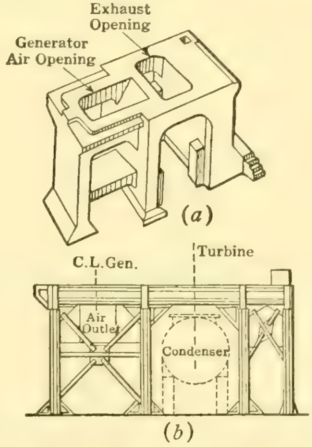


Fig. 304.

provided. The shells of the main bearings are sometimes arranged for *water-cooling*, but usually reliance is placed on the circulation of a copious supply of oil to carry away the heat generated at these points. As the boiler feedwater is used for cooling the oil, this heat is largely recovered.

(**h**) The **foundations** for turbines are usually of concrete, reinforced concrete, or structural steel, and are generally so arranged as to permit the condensers and various related auxiliaries to be placed directly under the prime mover, thus forming a compact unit which occupies a minimum of space. Typical turbine foundations are shown in Fig. 304 (*a*) and (*b*). The alignment and adjustment of turbines must be made with great care and must be maintained accurate at all times.

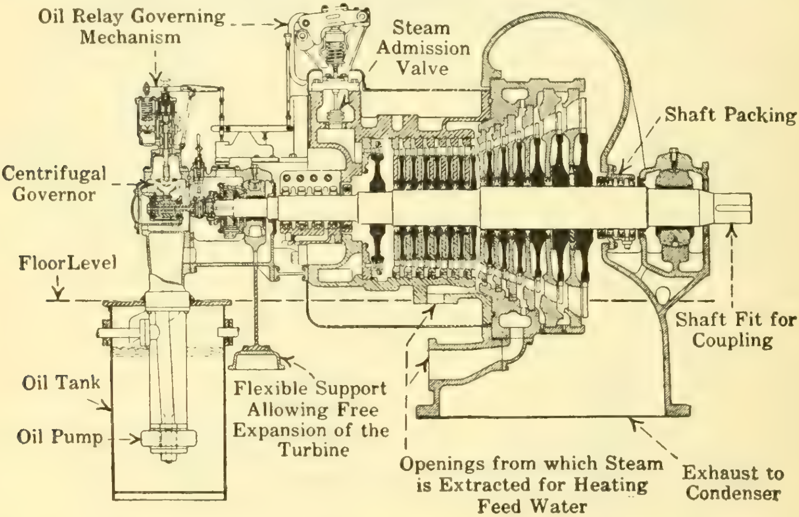

Oil Relay Governing Mechanism
Steam Admission Valve
Shaft Packing
Centrifugal Governor
FloorLevel
Shaft Fit for Coupling
Oil Tank
Flexible Support Allowing Free Expansion of the Turbine
Oil Pump
Openings from which Steam is Extracted for Heating Feed Water
Exhaust to Condenser

FIG. 305.—Sectional View of a General Electric Turbine of 2500-Kw. Rating, Showing the Arrangement of Wheels, Diaphragms, Bearings, Packing and other Features.

265. Illustrations of Medium and Large-Size Single Cylinder Turbines.—(a) Only a limited number of examples of a few makes of the larger turbines can be shown here, and in connection with them it must be remembered that there are other firms than those mentioned that make competing machines, and that most of the manufacturers have more than one type of design to select from to meet various conditions. In view of the preceding discussion the figures shown should be largely self-explanatory.

(**b**) Examples of medium and large-size **impulse turbines** are given in Figs. 305 and 306. The former shows a sectional view of one arrangement of medium-size General Electric turbine, and on it many of the parts are labeled. The first stage is velocity-compounded, the other stages having simple impulse blading. The machine is governed through a relay system with oil-operated gear which cuts the nozzles in and out of action to conform with the loading. Fig. 306 illustrates one form of large turbine of the same make. This machine has full peripheral admission in all stages, including the first, and has simple impulse blading throughout. Like all the other large turbines shown, this one has several openings from which steam can be extracted for heating the feedwater, or for other purposes. A moderate-size extraction turbine of the same make is shown in Fig. 322.

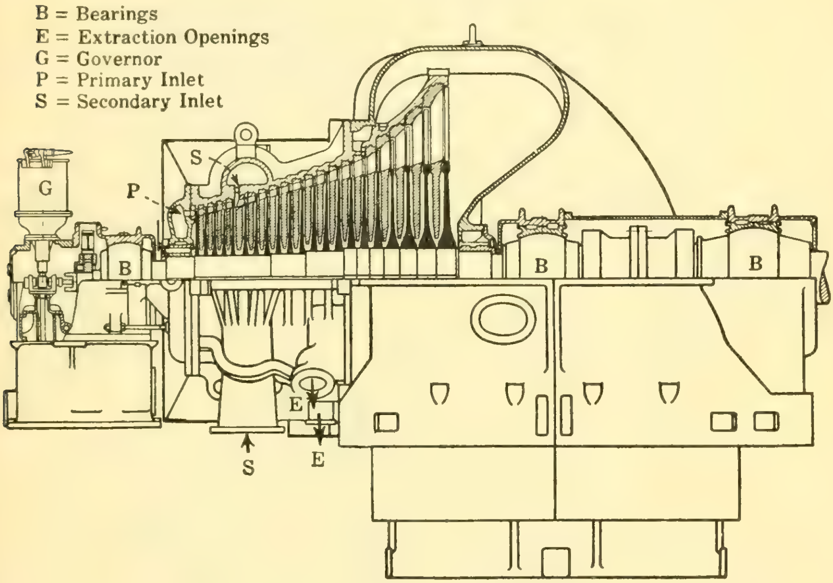


Fig. 306.—Sectional View of a 45,000-Kw. General Electric Turbine.

(c) Examples of moderate, medium, and large-size **combination type** turbines having velocity-compounding in the first stage combined with reaction blading in the other stages are given, respectively, in Figs. 307, 308, and 309. The first figure shows a bleeder turbine which can be transformed into a straight condensing machine by removing the intercepting bleeder valve and its operating devices, and blanking off the bleeder opening. The spindle is made in one piece and has no equilibrium piston, all end thrust being taken by the Kingsbury bearing. The main governor is of the centrifugal type and it regulates the oil pressure acting on the valve-operating mechanism shown at *A*. The larger Westinghouse turbines of this type, illustrated in Figs. 308 and 309, use a hydro-dynamic or impeller type of governor in which an oil impeller on the main shaft automatically varies, in accordance with the

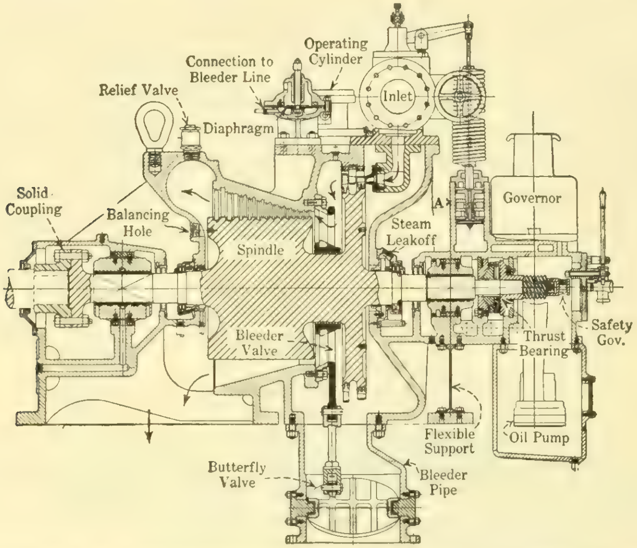


Fig. 307.—1500-Kw. Westinghouse Combination Type of Turbine, Arranged for Bleeding.

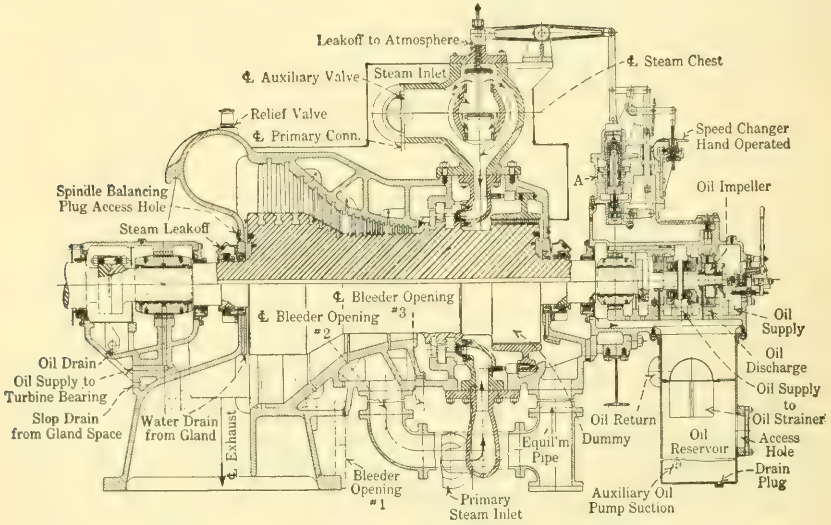


Fig. 308.—6000-Kw. Combination Type Turbine.

changes in turbine speed, the oil pressure acting on the relay piston. The impeller also supplies oil to the bearings. These turbines have dummy pistons and solid spindles with central inspection holes, the larger spindle being made in two parts.

(d) A combination of simple impulse blading in the first stages with reaction elements in the remainder of the machine is used in one form of turbine made by Brown-Boveri Company. One special form of turbine made by this firm is shown later, Fig. 319, the low pressure reaction blading being mounted on discs instead of drums, and the stationary blades having overlapping shroud rings. **Straight reaction** turbines are made in this country by the Allis-Chalmers Company, and one example of such a machine (using divided flow) is given in Fig. 312. In this turbine the rotor is built up, part of the drum being formed by the flanges of a series of discs.

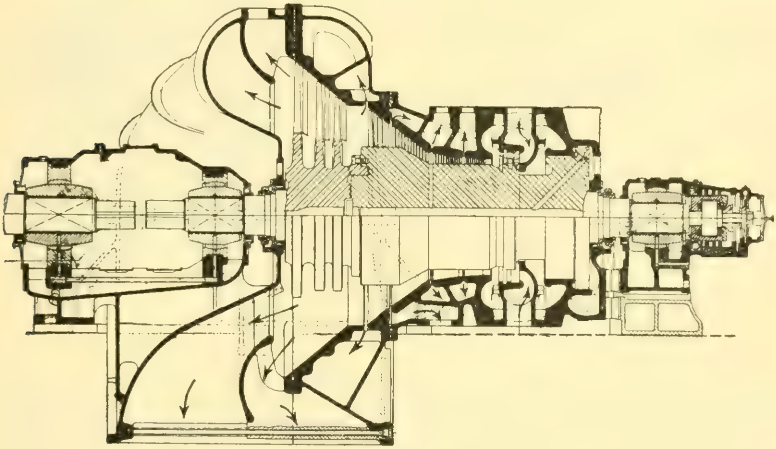
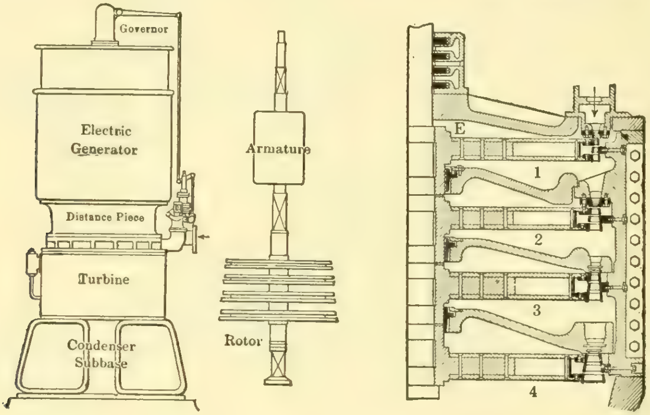


Fig. 309.—60,000-Kw. Westinghouse Turbine.

(e) Suggested **specifications** for large turbines and instructions to bidders are contained in the 1927 Proceedings of the National Electric Light Association, p. 853.

266. Special Arrangements of Turbines.—(a) Many **vertical-shaft** Curtis turbines, such as that illustrated in Figs. 310 (*a*) and (*b*), were formerly built in sizes up to 30,000 kw. These were the first large impulse turbines used, and many of them are still in service. All of the later large machines are of the horizontal arrangement which, in addition to having certain other desirable features, permits the use of higher rotative speeds that result in smaller and less costly turbines and generators for the same output than was possible with the vertical arrangement. Small-size turbines, however, are sometimes placed on the vertical shafts of pumps or similar machines.

(b) The specific volume of steam increases very rapidly as a high degree of vacuum is approached;[39] hence the use of very low back pressures, so conducive to thermal economy, involves the very serious difficulty of providing in large turbines adequate passages for the enormous volumes of vapor to be handled in the last stages of the machines. After reaching the maximum length of blade fixed by structural strength, and after increasing the relative discharge angle as much as is expedient (thus, however, augmenting the leaving loss, $v_2^2/2g$) some form of **multiple-flow** must be used with large units in order to provide adequate passages for the low-pressure steam. Some of the special arrangements adopted will now be considered.



a *b*

FIG. 310.—Vertical-Shaft Curtis Turbine.

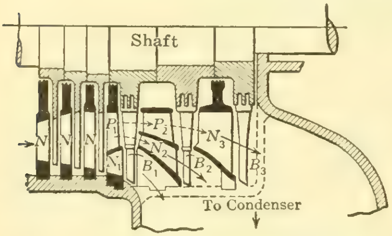


FIG. 311.—Baumann Divided-Flow.

(c) For the last row of blades that would normally be used in a single-cylinder turbine, the **Baumann divided-flow blading,** which is illustrated in Fig. 311, is sometimes substituted. In this figure N_1, N_2 and N_3 are nozzles or stationary expansion blades, the steam from which acts on the portions B_1, B_2 and B_3 of the rotating blades. The inner portions of the first two blades, and the stationary passages P_1 and P_2, conduct the steam to the succeeding nozzles; thus the flow in the last stages is divided into three streams, in the example shown, with the result that the blades used are shorter and of stronger construction than if the usual single row had been adopted. Other arrangements using multiple streams generally have blades of the usual form.

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[39] For example, the specific volumes at absolute pressures of 2, 1 and 0.5 in Hg. are approximately 330, 550 and 1250 cu. ft. per lb., respectively.

(d) An arrangement using divided **parallel double-flow** in the same direction is shown in Fig. 312, which illustrates one form of large reaction turbine made by the Allis-Chalmers Company. The **opposed double-flow arrangement** used by various manufacturers is illustrated in Fig. 313. In reaction turbines the latter arrangement automatically balances the end thrust. Turbines using a single type of element throughout, and those of the combination types, are sometimes made **semi-double-flow,** as in Fig. 314.

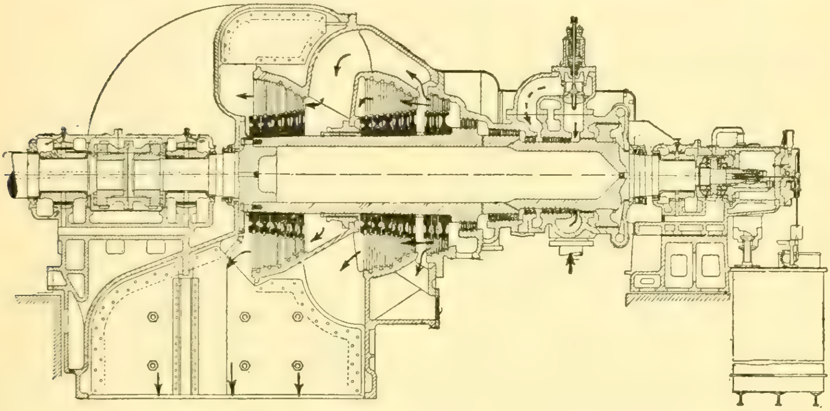

Fig. 312.—Parallel Double-Flow. (Allis-Chalmers.)

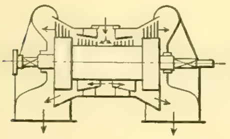

Fig. 313.—Opposed Double-Flow.

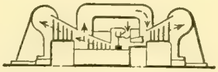

Fig. 314.—Semi-Double-Flow.

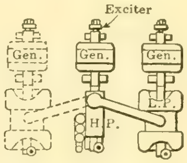

Fig. 315.—Cross-Compound Arrangements.

(e) Very large turbines may have **multi-cylinder arrangements,** the high-pressure and low-pressure elements being in separate cylinders to provide adequately for double flow of low-pressure steam and at the same time shorten the rotors and allow each to operate at its most suitable speed. The full lines in Fig. 315 show a two-cylinder arrangement of this kind. Very large turbines sometimes use two low-pressure elements, the second one being shown dotted in the last figure. These combinations are often called **cross-compound** units. Fig. 316 shows the arrangement of the 208,000-kw. multi-cylinder turbine in the State Line Plant. In the *vertical-compound* arrangement of the General Electric Company, illustrated in Fig. 317, the high-pressure turbine is placed on top of the low-pressure generator in order to save floor space, also one turbine foundation, and piping.

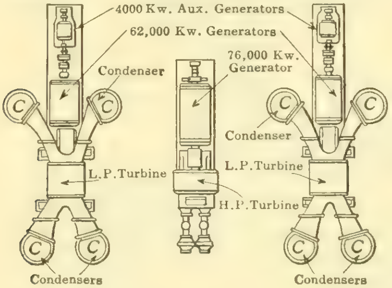

Fig. 316.—208,000-Kw. Unit (State Line).

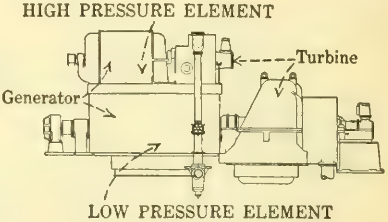

Fig. 317.—Vertical-Compound Arrangement (G. E.).

A two-cylinder **tandem-compound** arrangement, with double flow in the low-pressure element, is shown in Fig. 318. Fig. 319 shows a special three-cylinder Brown-Boveri tandem turbine in which the end thrust is neutralized by having the steam flow in opposite directions in the high- and intermediate-pressure cylinders, and by using opposed flow in the low-pressure element.

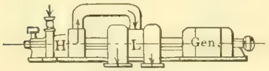

Fig. 318.—Tandem-Compound Arrangement.

(f) In the turbines so far considered the general direction of the steam flow is parallel to the axis of the shaft, hence the machines are said to have **axial flow.** Turbines using **radial flow,** usually with two reaction rotors that revolve in opposite directions, have been developed in

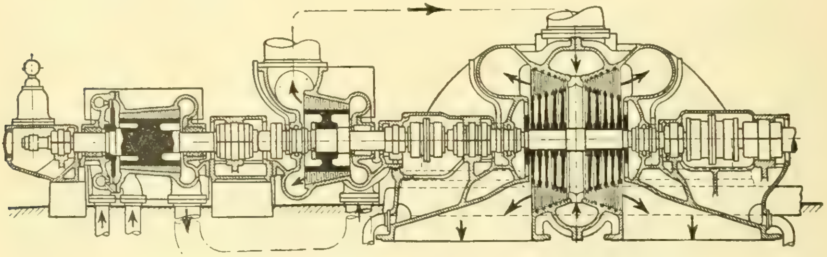

Fig. 319.—Three-Cylinder Tandem Brown-Boveri Turbine.

Europe by Ljungström. Such a turbine, the simple elements of which are illustrated in Fig. 320, is of a construction that is quite complicated and delicate; but, because of the double action and high relative speeds used, it is small when compared to the usual form of reaction machine of the same power.

In this country small radial flow impulse turbines, with various forms of velocity-compounding, are sometimes used to supply electricity for locomotive head-lights, search-lights, etc.

(g) **Low-pressure turbines,** one form of which is the low-pressure part of a multi-cylinder arrangement of turbine, are sometimes also used to receive steam from industrial sources that always have an adequate supply; and sometimes they utilize the exhaust from high-pressure steam engines. Under certain conditions reciprocating engines are able to use high-pressure steam somewhat more efficiently than do the first stages of a turbine; hence under these circumstances the latter arrangement may have a thermal advantage. The space occupied and initial cost of the combination, however, are usually prohibitive items, unless there are already existing engines which would otherwise be discarded and unless there is also space available for adding the turbines.

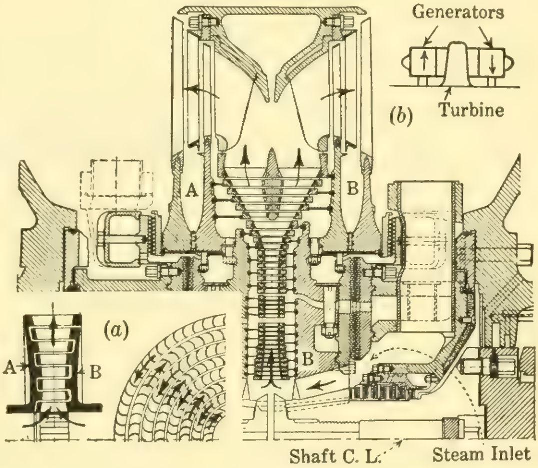

Fig. 320.—Ljungström Radial Flow Combined with Low-Pressure Axial Flow.

(h) Turbine elements using steam at very high pressures, such as 1200 lb. per sq. in. or more, with high temperatures, require unusually careful designing and the use of special materials and construction; hence, the stages using steam at such pressures are commonly segregated and made to constitute a machine separate from the rest, and this part is referred to as a **high-pressure turbine** or cylinder. When economic conditions warrant, such turbines are combined in new installations with low-pressure condensing elements to make complete units; sometimes they are added, with the necessary high-pressure boilers, to existing moderate-pressure systems to increase the capacity and efficiency thereof, the high-pressure exhaust (after re-superheating) being delivered to the original turbines along with the superheated steam from the original boilers; and sometimes high-pressure turbines are used in industrial plants in which their exhaust is utilized as a heat carrier at high pressure. In the latter case the machine is sometimes called a **back-pressure turbine.** One illustration of a high-pressure impulse turbine which receives steam at 1200 lb. per sq. in. and discharges it at about 350 lb. is given in Fig. 321, the blade wheels of the machine being made solid with the shaft. This turbine and those receiving steam from it at the lower pressure carry with excellent economy the rather low base load of the plant in which they are installed, the fluctuations in load being met by additional 350-lb. turbines that receive steam from boilers serving them at that pressure. Before being delivered to the low-pressure turbines the exhaust from the high-pressure element is reheated.

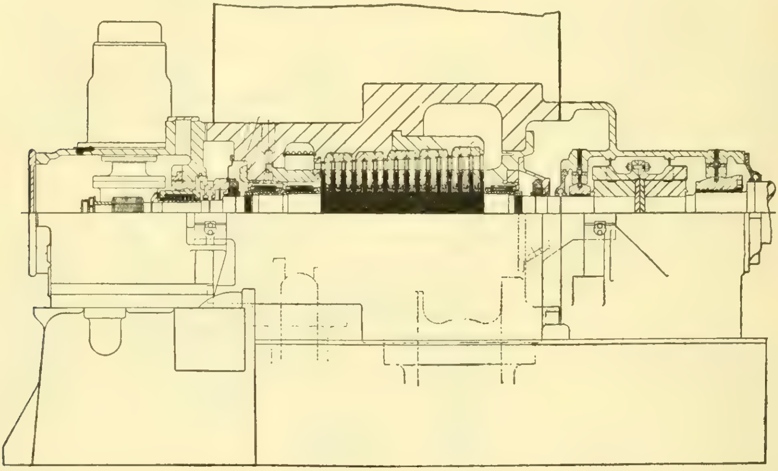

Fig. 321.—Section of 10,000-Kw., 3600-R.p.m., 16-Stage, 1200-Lb., General Electric Company Turbine. (Edison Illum. Co., Boston.)

(i) **Reheating turbines** are those in which the steam, after passing through part of the machine, is reheated by some external source of heat, such as the hot gases from a boiler or high-pressure steam, before it is used in the low-pressure element of the machine. As the reheater and its piping may contain enough steam to run the turbine for a considerable length of time under a light load, some arrangement is desirable to prevent this steam from causing the turbine to run away after the main safety trip has functioned in an emergency. The chief *advantages* gained by reheating the steam are:

(1) The engine efficiency is increased because the moisture in the low-pressure stages is eliminated or greatly reduced, thus increasing their stage efficiencies.
(2) The maintenance cost of the turbine is reduced because the erosion of the turbine blades due to moisture is much less.
(3) The ideal reheating cycle has a slightly higher efficiency than the Rankine, as was shown in Chapter XVIII, Part I; and this fact, coupled with the higher engine efficiency, assures a higher thermal efficiency of the turbine.

The *disadvantages* of the reheating turbine are those of extra first cost, and additional space. The advantages become more pronounced as the boiler pressure is increased, so that for pressures of 600 lb. per sq. in., or more, the reheating equipment is well justified if the yearly load factor is fairly high. The optimum reheating pressure is considered in the next chapter, pages 117 and 118.

(j) **Marine turbines** which are on the same shaft with their propellers should be operated at speeds which give maximum efficiencies for the latter, or for the combination, in which case, on slow ships, their rotative speeds are relatively low and hence the turbines are large. However, this difficulty may be surmounted by driving through mechanical reduction gears, that are usually of the herring-bone type, which may have high mechanical efficiencies (98+ per cent). Turbines driving directly or through gears are provided with a reversing element which is incorporated within them or is in a separate casing; and such turbines are quite inefficient except when operating at their most economic speeds, which should correspond with the normal speed of the ship. Flexible operation, with the maintenance of constant turbine speeds and more economical operation, may be accomplished through the use of hydraulic drives (such as the Föttinger), or, more commonly, through electric motor propulsion, the current being supplied by turbine-generator units which are somewhat similar to those used in stationary power plants. The geared arrangement is, of course, the least costly and most commonly used.

(k) The special arrangements and applications of bleeder and mixed-pressure turbines involve modifications of construction and will be considered in the next section.

267. Bleeder and Mixed-Pressure Turbines.—(a) In industrial plants, steam at low or medium pressure is often required for heating buildings, drying, cooking, chemical processes, or other purposes. When there is also a demand for energy to drive machines, the manufacturer has several optional methods of meeting the requirements. One is to purchase the needed electrical energy from a central station, and to install boilers for supplying the steam for heating. Another is for the manufacturer to generate his own power and obtain the heating-steam directly from boilers. But usually the preferred method is to generate the steam at a much higher pressure than that required for the heating purposes, and use in the prime mover as much of its heat as is possible before delivering the steam to the heating system. In the latter case the heat is utilized with the maximum efficiency, the power generated being commonly termed *by-product power.*[40] If the energy requirements for power and heating were always equal, as would be desirable, then a *back-pressure* prime mover could be used and all of its exhaust could be applied for the heating; but, unfortunately, the power and heating loads are seldom equal, and usually they are variable, often being seasonal. The conditions sometimes may demand the full power output when there is little or no call for low-pressure steam, and at other times require much steam for heating and little for power.

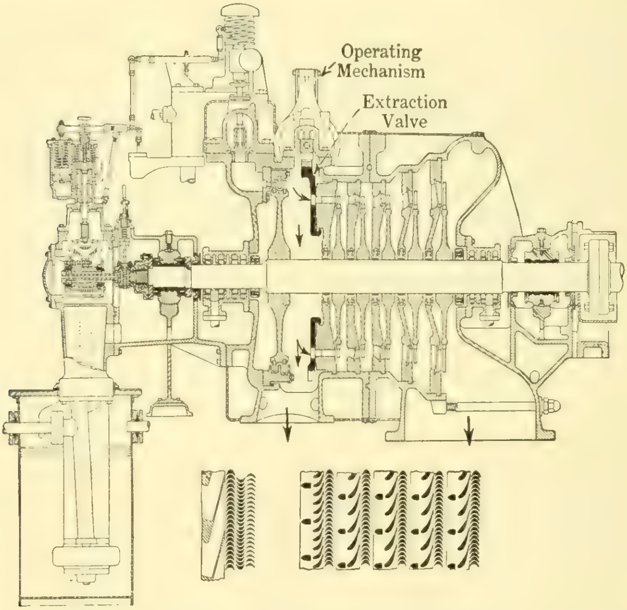

FIG. 322.—One Form of General Electric Bleeder Turbine.

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[40] The energy consumption in B.t.u. per kw-hr. of generator output in this case is equal to the ideal rate (3413) corrected for the engine efficiency of the unit, as compared with 9000 to 12,000 B.t.u. per kw-hr. with some of the best large central station units.

(b) **Bleeder turbines** meet the need for a type of prime mover that automatically adjusts itself to wide ranges of power and heating demands. In the simplest form of such a machine there are high- and low-pressure sections with a bleeding point between. When heating-steam is required at two or more pressures or temperatures, the different demands are met by bleeding from the several appropriate stages of the turbine. Bleeder turbines are usually condensing, although conditions may sometimes require one that is non-condensing, and they generally have regulating devices to control the outflow of the steam bled. The distinction between a *regenerative* extraction-turbine and the usual *bleeder turbine* is primarily in the use of the extracted steam. The steam bled from the first is utilized to heat the feedwater, thereby increasing the thermal efficiency of the plant, and no pressure regulators are required in the extraction lines, as the rate of condensation of the steam in the feed-water heaters automatically controls the flow of steam thereto; whereas the bleeder turbine furnishes steam for heating or industrial processes, and the flow of such steam must usually be controlled by some form of regulator.

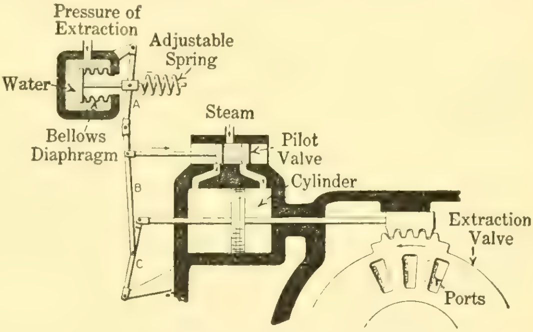

Fig. 323.—Elements of a Pressure Regulator.

(c) One arrangement of bleeder turbine is shown in Fig. 322, and another in Fig. 307, the main elements being similar to those of the more usual forms of turbines. If the extraction is to be at fixed pressure regardless of the throttling of the steam supplied to the turbine, then the pressure-regulator, one arrangement of which is shown diagrammatically in Fig. 323, commonly consists of some kind of pressure-actuated spring-balanced diaphragm or piston which controls (directly in small turbines, or through a relay system on large machines) an intercepting, bleeding or extraction valve, often in the form of a grid, which is placed between the point of extraction and the next succeeding stage. If the steam pressure at the bleeding point is too high, the control valve allows more steam to flow through the succeeding stages of the unit, and if too low the valve restricts the flow to those stages.

If the turbine supplies extraction steam to a pipe line into which other units are also bled, some safety device, such as the automatically controlled butterfly valve in Fig. 307, may be needed to prevent the steam from the other turbines from driving the one under consideration and causing it to run away when lightly loaded. The high-pressure element of a bleeder turbine differs from that in the usual turbine because it must be made large enough to generate by itself all the power required, assuming that the full electrical demand must be met at the same time that the maximum bleeding occurs. Under light loading the high-pressure element of such a turbine is therefore less efficient than that in a standard machine. If the maximum bleeding is but a part of the total steam entering the turbine, or if the biggest extraction never synchronizes with the greatest power demand, the nozzle and blade passages in the high-pressure stages can be correspondingly reduced, and the efficiency of this part is thereby increased. Never should *all* the steam be extracted, because it is always necessary to have a sufficient amount of *ventilating steam* pass through the low-pressure element to prevent the blades and wheels therein from being damaged by the heat generated by their windage in the stagnant vapor. Obviously, a bleeder unit should be proportioned to fit the special conditions under which it is to function.

(d) When by-product steam from industrial processes, steam hammers, pumps, old non-condensing engines, or other sources is available in adequate quantities it can be used for producing power without entailing any expense for fuel. Usually, however, the power is also required at times when the supply of by-product steam is insufficient or not available, and then the deficiency may be overcome by using high-pressure steam. A **mixed pressure turbine** is one designed to meet these varying conditions in a single machine. It has a low-pressure element for utilizing the by-product steam, but also is provided with a high-pressure section to which live steam is automatically admitted by its governor when the normal supply is inadequate to meet the power demand. In such a turbine the passages in the low-pressure element are necessarily much larger than those in the corresponding standard machine of the same capacity; hence, if such a turbine were operated entirely with high-pressure steam, its performance would be somewhat poorer than that of the corresponding machine of the ordinary type.

(e) In some industries there is at times an excess of low-pressure steam available for generating power, whereas at other times there is a deficiency of process steam which might be overcome by extracting steam from a turbine. **Combined mixed-pressure and bleeder turbines** are designed to meet such conditions with a single machine that functions automatically as either type of unit.

In other types of bleeder or mixed-pressure turbines a large proportion of the steam extracted from the high-pressure stage is still available after doing its process work, and this steam is returned to the lower stages of the turbine to do additional work therein.

268. Steam Turbine Troubles.—(a) The steam turbine stands supreme today in the large fuel-burning central station because of the small space occupied, light foundations, low first cost, high thermal economy, and reliability. However, it is a machine that has to be carefully designed, manufactured, erected, and operated in order to give satisfactory service.

(b) One of the most common causes of "*outage*," or the removal of a turbine from service, is that some part of the *rotor has made contact with a stationary part.* Since the wheel rim of a turbine runs at a very high speed any contact that it makes with the stationary portion is likely to cause serious trouble and may even destroy the machine. In the early turbines the designers and operators did not always pay sufficient attention to the relative expansion of the casing and rotor and consequently serious rubbing often occurred when the load was suddenly changed. Also, many early turbines were designed without proper consideration being given to the distortion of the casing produced by the expansion of the attached steam pipe.

(c) The *blades* of the last stages that handle wet steam are subject to very rapid erosion due to the moisture. On the other hand, the first stage buckets may be subjected to very high temperature steam, which may soon cause them to fail. Aluminum bronze, cartridge brass, nickel brass, delta metal, phosphor bronze, manganese bronze, nickel steel, stainless steel, nickel stainless steel, Monel metal, and nickel have been used for blades, but no material thus far developed can be said to be entirely satisfactory to meet all the severe requirements of our modern large turbines. In addition to moisture and high temperature, which have already been mentioned, turbine blading may fail because of bucket vibrations, mechanical abrasion from boiler scale, chemically impure steam, mud deposits, rusting during idle periods, defective material, or embrittlement. One illustration of the difficulty of securing a satisfactory material for blades will be mentioned: A particular type of stainless steel frequently fails from vibration, yet it has been found to be free from chemical attack, has high tensile strength, and is in most respects very satisfactory. The failure of one blade may cause the destruction of several rows of buckets or even a complete wreck of the turbine.

(d) The *main shaft* of a turbine may fail due to a flaw in the material, or due to vibration and reversal of stresses, or a combination of these factors. However, this type of failure seldom occurs but it is always a possibility and the larger the machine the more serious it is as to possible loss of life, destruction of property, reduction of plant capacity, and the time and money required to rebuild the machine.

(e) Turbines sometimes fail because of trouble with the *lubrication system, bearings, diaphragms, packing* or *governor*, but the records [41] of 207 large units, having a total capacity of 6,768,900 kw., show that the turbines caused their own outage only 5.44 per cent of their total hours during the year 1928. This and many other records prove that the modern large steam turbine is a very reliable machine, but its annual inspection and adjustment as well as its daily operation require painstaking care. Small turbines generally require considerable less attention and care than do large ones.

(f) The **fire hazard** occasioned by the use of steam having the high temperatures now commonly employed with large turbines, has recently become of prime importance, as several disastrous fires have resulted from the leakage of oil near hot steam parts of such units. Careful attention should be given to the design, attachment, and location of the piping supplying oil for lubricating the bearings and for operating the control gear of such units; provision must also be made for the safe storage of this oil. Eventually some suitable non-combustible liquid having the requisite lubricating qualities may be discovered and substituted for the combustible oil now employed.

ADDITIONAL REFERENCES

"High-Pressure Boiler and Turbine Operation at Northeast Station," by J. A. Keeth, Trans. A.S.M.E., Aug. 1932, FSP-54-13, p. 161.

"Steam-Turbine Plant Practice in the United States," by V. E. Alden and W. H. Balcke, Trans. A.S.M.E., March 1933, FSP-55-3a, p. 9.

"British Practice in Steam-Turbine Design," by F. W. Gardner, Trans. A.S.M.E., March 1933, FSP-55-36, p. 37.

"The Thermal Performance of the Detroit Turbine Using Steam at 1000° F," by W. A. Carter and F. O. Ellenwood, Trans. A.S.M.E., July 1934, FSP-56-8, p. 477.

"High-Temperature Steam Experience at Detroit," by P. W. Thompson and R. M. Van Duzer, Jr., Trans. A.S.M.E., July 1934, FSP-56-9, p. 497.

"Leaving Velocity and Exhaust Loss in Steam Turbines," by E. L. Robinson, Trans. A.S.M.E., July 1934, FSP-56-10, p. 515.

"50,000 kw. Ljungstrom Turbo-Generator," Engineering, July 6, 1934, p. 1.

"Reducing Turbine Fire Hazards," in Combustion, Sept. 1934, p. 33.

"High-Pressure-Steam and Binary Cycles as a Means of Improving Power-Station Efficiency, by G. A. Gaffert, Trans. A.S.M.E., Oct. 1934, FSP-56-11, p. 755.

"Dynamic Balancing of Rotating Machinery in the Field," by E. L. Thearle, Trans. A.S.M.E., Oct. 1934, APM-56-19, p. 745.

"Recent Developments in Steam Turbines," by C. R. Soderberg, Mech. Eng., March 1935, p. 165.

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[41] See the Nat. El. Lt. Assoc. Prime Movers Committee Report, Aug., 1929, p. 20.

CHAPTER XXII

PERFORMANCE OF STEAM PRIME MOVERS AND PLANTS; AND SELECTION OF UNITS

269. General Requirements.—(**a**) As a supplement to the *thermal performance* of the unit, which is always a factor of more or less importance in the selection of the type and size of steam engine or turbine for a given installation, good engineering also requires that careful attention be given to the following **additional considerations:**

(1) *Local conditions,* such as the most suitable steam pressure at the throttle, the most satisfactory exhaust pressure, the water supply available for boiler feedwater and use in the condenser, the possible utilization of bled or exhaust steam, the variable load characteristics, and the daily and seasonal variations in demands for power and steam.
(2) *First cost of complete unit,* including auxiliaries.
(3) *Cost of buildings* and *foundations.*
(4) *Operating expenses* for fuel, attendance, supplies, and repairs.
(5) *Overhead charges* for maintenance, depreciation, taxes, and interest on the investment.
(6) *Influence on the selection of other equipment,* such as boilers, condenser, piping, and the generator, or other connected machine.

The discussion of these general economic factors will be deferred until later, the present chapter being devoted primarily to the consideration of the thermal performances of the various types of steam prime movers and of the plants using them.

(**b**) The **steam turbine** is inherently a high-speed machine. It is applicable for the direct driving of electric generators, centrifugal pumps and compressors, and other high-speed apparatus; and, through the very efficient reduction gearing that is now available, it is also widely used for driving low-speed machinery. A turbine costs less and requires smaller space, lighter foundations, and less attendance than does a reciprocating steam engine of the same output, and, further, is without vibrations that may be produced by reciprocating masses. Since the turbine has no internal lubrication, steam exhausted or bled from it is free from oil that would be undesirable to have deposited on the heat transmitting surfaces of condensers, and boilers, or in heaters used in various industrial processes. Under favorable conditions as to vacuum, initial pressure, and superheat, turbines of moderate cost and size (say 1000–3000 kw.) have thermal performances equaling those of steam engines of the same capacities. For very large amounts of power, such as 30,000 kw. and above, turbine units are the only ones feasible to construct, except for very special conditions, and they also give better thermal economies than would be possible with reciprocating units. Small non-condensing turbines, although having poorer thermal performances than engines of the same size, may nevertheless be preferred in some cases because of their smaller cost, and greater convenience, and they are especially applicable when all of the exhaust steam is to be utilized and must be free from oil.

(**c**) The **reciprocating engine** is superior to the turbine when the operation must be with slow rotative speeds, variable speeds, or reversing; and also when thermal economy is important in a small unit. In sizes less than about 500 kw. its steam consumption is generally less than that of the turbine, especially if the operation is non-condensing. Under favorable conditions, condensing uniflow and poppet-valve engines have usually shown better thermal economies than turbines of few stages even in sizes up to 3000 hp. or above, but the very much lower cost of this type of turbine, combined with its small space and high speed, often gives it the advantage, especially if used for driving an electric generator.

For ordinary steam pressures, say those below 300 lb. per sq. in., the future field of application of the piston engine will probably be very largely confined to locomotives, rolling mills, reciprocating pumps and compressors, mine hoisting equipment, driving small generators and certain types of marine apparatus, and to moderate-size installations requiring good thermal economy. The last includes the many applications in which the demand for exhaust steam for use in heating is less than the amount of steam supplied for generating the power needed, or those in which the heating is seasonal. If the oil in the exhaust steam from an engine is objectionable, it can be removed quite effectively by means of an oil separator. For very high steam pressures at the throttle and with exhaust pressures considerably above atmospheric some engineers believe that the use of the piston engine may be considerably enlarged, because it may prove to be superior to the turbine in this field.[1] Many tests[2] of compound steam engines show the indicated engine efficiency to be much better in the high-pressure cylinder than in the low; whereas with the turbine, especially when very high pressures are used in small units, the leakage in the first stages may easily reduce the efficiencies there to values considerably less than those in the lower pressure range, and to less than those in the high-pressure part of a reciprocating engine.

270. Methods of Comparing Performances of Steam Prime Movers.—(**a**) The comparison of engine or turbine performances by means of their **steam rates** became common in past years and, unfortunately, the practice is often still continued by some engineers when there is no justification for it. Steam rates are accurate measures of comparing performances of different prime movers only when the units have:

(1) The same state of the steam at the throttle.
(2) The same exhaust pressure.
(3) The same cycle.

If there is only a slight variation in items (1) and (2), **correction factors** may be introduced to make the comparison of the *steam rates* a fair one. These factors should be based on accurate data showing the effect of slight changes in the throttle pressures, throttle temperatures, and exhaust pressures of the particular units under consideration. If values of item (1) or (2) are widely different, or if the engine cycles are not the same, the steam rates should not be used as comparative measures of performance; nor are they, or even the thermal efficiencies, proper criteria unless the units are of about the *same size*, since, generally speaking, large units may always be expected to be more efficient than small ones, as will be shown later.

(**b**) The **engine efficiencies,** on the other hand, are far better guides to the performance of steam prime movers in general because they are inherently the only means of measuring the portion of available energy utilized, and are not so sensitive to small changes in exhaust pressure and admission pressure as are the steam rates. Since the different engine efficiencies are so important to keep clearly in mind, Eqs. (185),

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[1] The Steam Engine as a High-Pressure Industrial Prime Mover, by John F. Ferguson, Power, May 28, 1929; also abstract of article by J. Kluitmann, Z.V.I., Vol. 71, No. 46, in Mech. Eng., Jan., 1928, p. 159.

[2] Schmidt Steam Engine More Efficient than Largest Turbine, by O. H. Hartman, Power, Feb. 7, 1922.

(186), and (187) from Chapter XII, Part I, will be repeated here, with new numbers for convenience. Thus,

$$\text{Indicated engine efficiency} = \frac{\text{Indicated thermal efficiency}}{\text{Ideal cycle efficiency}}. \tag{370}$$

$$\begin{array}{l}\text{Brake engine efficiency}\\ \text{(or the engine efficiency)}\end{array} = \frac{\text{Brake thermal efficiency}}{\text{Ideal cycle efficiency}}. \tag{371}$$

$$\begin{array}{l}\text{Combined engine effi-}\\ \text{ciency (or the engine effi-}\\ \text{ciency of the unit)}\end{array} = \frac{\text{Thermal efficiency of combined unit}}{\text{Ideal cycle efficiency}}. \tag{372}$$

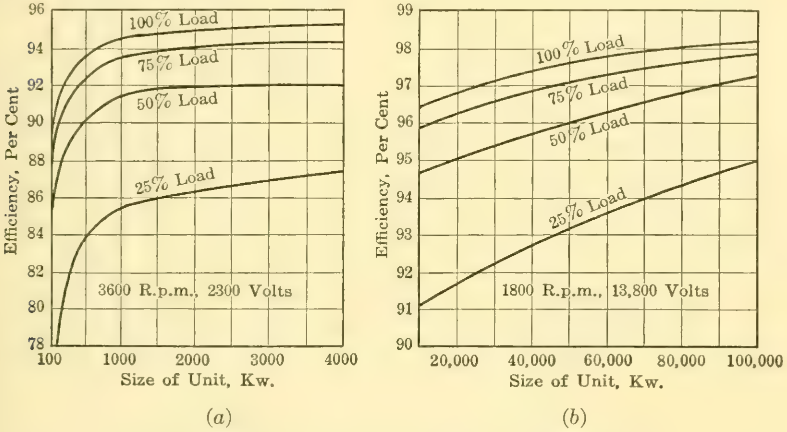
If a piston engine operates on the Rankine cycle the *indicated engine efficiency* is also equal to the ideal steam rate in lb. per hp-hr. divided by the actual steam required per i.hp-hr. Clearly this efficiency cannot be used for steam turbines because the *indicated* horsepower of a turbine cannot be obtained. For a reciprocating steam engine, however, this efficiency can be determined easily and is of value in showing what portion of the energy available from the steam in passing through the engine has been delivered to the piston. This efficiency measures the degree of success attained in eliminating leakage, initial condensation, throttling, radiation and incomplete expansion, but the mechanical losses are not involved.

Unfortunately, too many reciprocating engine tests have been reported in the technical press without giving the results in such manner that the *brake engine efficiency* can also be found. The latter is the real measure of performance generally desired, because the *delivered power* is an item of far more importance to the owner of the engine than the *indicated* power. Mechanical efficiencies of different engines vary considerably and their values cannot, in general, be estimated accurately enough to be useful in calculating the brake engine efficiency from the indicated value.

If a piston engine or a turbine operates without bleeding or reheating, and only in that case, its *brake engine efficiency*, as given by Eq. (371), is also equal to the ideal steam rate divided by the actual steam used per brake horsepower-hour.

The *combined engine efficiency*, Eq. (372), of a unit consisting of the prime mover and its connected machine may be found, when there is no bleeding or reheating, by dividing the ideal steam rate by the actual steam rate measured on the basis of the output of the driven machine. In such units the driven machine is generally an electric generator, and in that case the combined engine efficiency becomes the ratio of the ideal steam rate in pounds per kilowatt-hour to the actual steam required per kilowatt-hour delivered by the generator. The combined engine efficiency takes into account all generator losses as well as all of the engine losses that differentiate the actual and ideal engines.[3] Clearly the *engine efficiency* of the engine or turbine alone, or the *turbine* efficiency as it may be called for the latter, is found by dividing the combined engine efficiency by the generator efficiency. For turbine-generator units for which the generator efficiency is not known, the values given in (**d**) of this section may be used as close approximations.

(**c**) The **combined thermal efficiency,** or *thermal efficiency of a unit,* is always the ratio of the energy delivered by the combination, to that supplied to it. This efficiency is a very important item, especially in connection with large power plants. The reciprocal of the thermal efficiency multiplied by 3413 gives the **energy consumption** of the unit in B.t.u. per kilowatt-hour.



(*a*) (*b*)

FIG. 324.—Efficiencies of A.C. Turbine-Driven Generators. (60-Cycle, 80 Per Cent P.F.)

(**d**) **The efficiencies of electric generators,** as already noted, are needed in determining the thermal performance of the *complete unit* from that of the prime mover alone, or vice versa. The manufacturers of the generators can always supply accurate values of these efficiencies for the different sizes of their machines under the various operating conditions. Fig. 324 gives representative efficiencies of alternating current generators of the types used with turbines. These curves cannot be considered as giving exact values for all cases, but they are reliable for most modern machines operating at the specified speeds, cycles, voltages, power factors, and loading percentages. It should be noted that the larger the generator and the more fully it is loaded, the better is the efficiency. For example, a 4000-kw. generator at *full* load has an efficiency no better than that of a 20,000-kw. machine at *half* load, or a 100,000-kw. one at *quarter* load.

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[3] The methods of finding the combined engine efficiencies and thermal efficiencies of units operating on the regenerative and reheating cycles are given in Chapter XVIII, Part I.

(e) **The Power Test Codes** prepared by the A.S.M.E. should be followed as far as possible in making commercial tests of steam engines or turbines.

271. Small Turbines and Reciprocating Engines.—(a) For most steam prime movers in capacities up to 2000 kw. the steam pressure used is relatively low, say 250 lb. per sq. in. or less, and the temperature of the steam is not high compared with that used in some of the latest large central stations. These small units nearly always operate without any regenerative feedwater heating and without reheating the steam, except in the compound reciprocating engine; consequently the *Rankine engine cycle,* as discussed in Sect. **203,** Part I, is properly chosen as the standard on which to base the engine efficiencies of these small turbines or engines.

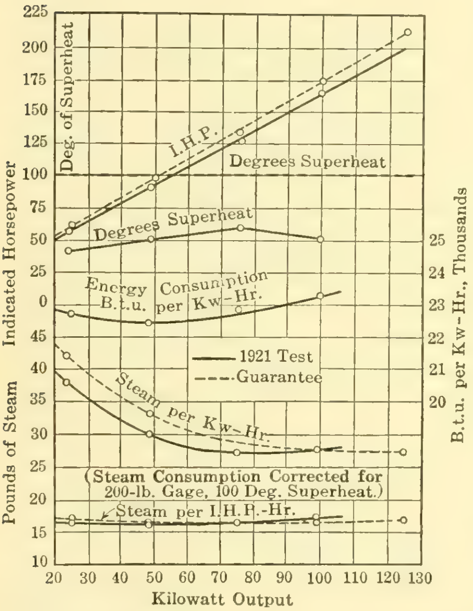

Fig. 325.—Small Uniflow Engine Performance Curves.

(b) An example of the performance of a 100-kw. *non-condensing uniflow engine* and generator (A.C.) is shown in Fig. 325. These results[4] were obtained at the Division Avenue Pumping Station, Cleveland, with an exhaust pressure of 2 lb. per sq. in. gage. The superheat and throttle pressure are shown in the figure. The steam consumption curve of this engine is clearly a remarkably *flat one* on the i.hp-hr.

[4] Power, Jan. 17, 1922, p. 98.

basis, but it is appreciably different when the generator efficiency and the mechanical efficiency of the engine are considered, as is seen from the curve for steam consumption per kilowatt-hour. The product of these two efficiencies, as determined from the two curves, is about 82 per cent at full load and 59 per cent at quarter load. The generator efficiency, with a power factor of 80 per cent, was known to be about 0.89 at full load and 0.78 at quarter load. The corresponding mechanical efficiencies of this engine at these loads are therefore found to be 0.92 and 0.76, respectively. The ideal steam rate for the steam pressure and superheat given in the figure, and for a back pressure of 2 lb. above atmosphere, is 17.1 lb. per kw-hr. Consequently the combined engine efficiency of this unit is about 62 per cent at full load and 44 per cent at quarter load. This is much better performance than can be expected of most small non-condensing turbine units.

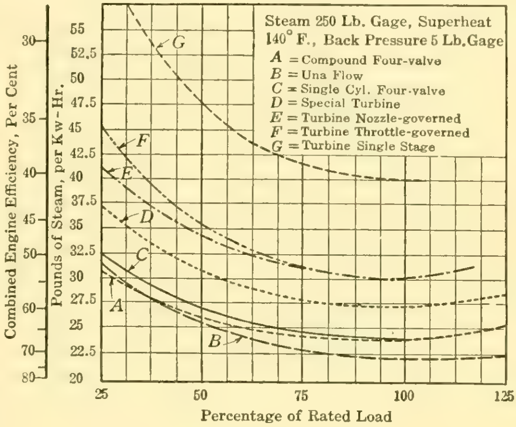

FIG. 326.—Comparative Steam Rates of Non-condensing Units in Sizes from 750 to 1500 Kw.

(c) The **comparative steam rates** of several different kinds of small non-condensing engine and turbine-generator units[5] in capacities from 750 to 1500 kw. are indicated in Fig. 326, being based on the assumption that the steam pressure is 250 lb. per sq. in. gage, with 140 deg. superheat, and that the exhaust pressure is 5 lb. above atmosphere in each case.

A scale by means of which the same curves can be used to show the combined engine efficiencies of these units has been added to the original figure, and from it the *uniflow unit* is seen to give an efficiency of about 72 per cent at full load and about 52 per cent at quarter load. The *single-stage turbine-generator unit*, on the other hand, gives a combined engine efficiency of only about 40 per cent at full load and about 26 per cent at quarter load. By increasing the number of stages in the turbine, however, more efficient results can be obtained, as shown by curves *D*, *E*, and *F* for the multi-stage turbines, but even then they are not equal to those of the engines under the conditions given.

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[5] Prime Movers for Industrial Plants, by John F. Ferguson, Power, May 13, 1924.

The flatness of curves A, B, C, and D indicates a very satisfactory range of load over which the variation in steam consumption is not large. The cost of the multi-stage turbines is, of course, greater than for the single-stage machine, but it is less than for the engines.

(d) The **first cost** of small turbine- or engine-generator units of given size varies almost directly as the combined engine efficiency of the unit at full load. For example, a uniflow unit of 500 kw. capacity will cost about \$50 per kilowatt and will have a combined engine efficiency of from 70 to 74 per cent, whereas the single-stage turbine unit will cost only about \$28 per kilowatt and its efficiency will be only from 40 to 50 per cent. The four-valve-engine unit will cost about \$42 per kilowatt and its efficiency will be about 66 per cent. In other words, it requires a considerable additional expenditure of money to build an engine or turbine of such construction that it can utilize the available energy of the steam with the minimum waste from cylinder condensation, leakage, friction, windage and leaving loss. Also engines or turbines of large output generally cost less per unit of capacity than do small ones, and the higher the speed the smaller will be the size and cost of the prime mover and its driven machine for a given output. Illustrative prices of turbine-generator units of various sizes and speeds are given in Fig. 327.[6]

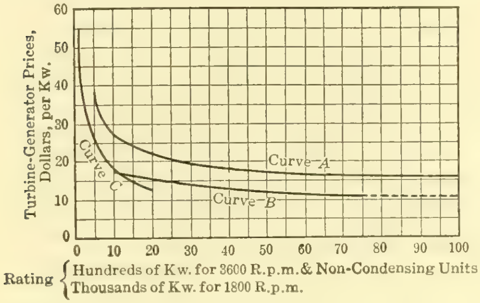

FIG. 327.—Prices for Turbine-Generator Units in the United States.

The prices given in the figure are for the bare turbine and generator, without auxiliary generators or direct-connected exciters, and are for units delivered and erected. They are for pressures up to 400 lb. per sq. in. gage, steam temperatures up to 725 deg. fahr., and when condensing, up to 29 in. vacuum.

Curve A—3600 r.p.m., Condensing Units.
Curve B—1800 r.p.m., Condensing Units.
Curve C—Small Non-Condensing Sets.

(e) The **number of stages** used in a turbine or the number of cylinders compounded in steam engines also has an important effect on both the performance and cost of the machine. Several turbines in sizes from 1000 to 6000 kw. which have recently been constructed with large numbers of stages, have shown very high efficiencies in their high-pressure sections. For example, a Bunner[7] 2400-kw. non-condensing turbine-generator unit having a steam pressure of 202 lb. per sq. in. abs. and a temperature of 741 deg. fahr. at the throttle, and with a back pressure of 22.2 lb. per sq. in. abs., gave a *combined* engine efficiency of 77.9 per cent at full load and 59.2 per cent at half load. For the turbine alone these figures correspond to engine efficiencies of 82 and 65 per cent, respectively. These excellent results are attained because the blade and steam velocities are reduced by using many stages; but the thermal gain is effected only with more expensive construction. Similarly, by making steam engines multiple expansion their performance is improved but at a greater cost for the machine.

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[6] From Economic Considerations in the Application of Modern Steam Turbines to Power Generation, by A. G. Christie, Mech. Eng., July, 1930, p. 673.

Manufacturers in this country are prepared to build small turbines with many stages and high efficiency,[8] but the high first cost of such machines limits their field of application to those cases having large yearly load factors and relatively expensive fuel.

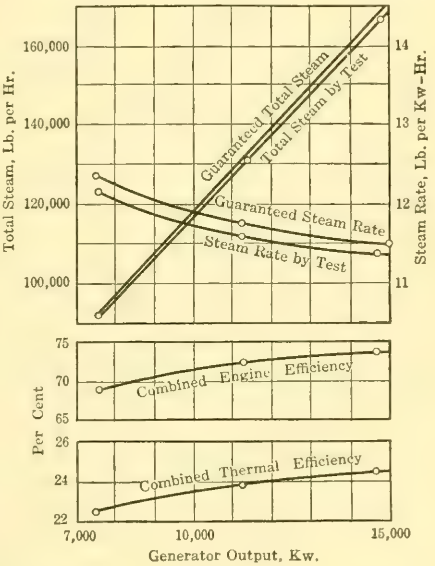

FIG. 328.—Test Results from a 15,000 Kw. Unit.

272. Turbine-Generator Units of Medium Size (Rankine Cycle).—(a) A good example of the performance of a 15,000-kw. unit is given by the test-results[9] shown in Fig. 328. This unit consists of a single-flow Allis-Chalmers reaction turbine, designed to operate at 1800 r.p.m. with a steam pressure of 225 lb. per sq. in. gage, 125 deg. of superheat and an exhaust pressure of 1 in. Hg abs. The turbine is directly connected to a 3-phase, 60-cycle, 11,000-volt generator. The *total consumption* curve is seen to be substantially straight, and it therefore conforms quite exactly to *Willans Law* (p. 414, Part I), which is a characteristic of most turbines up to the load at which the secondary valve begins to open. The combined engine efficiency of this unit is shown to vary from 69 per cent at half load to 74 per cent at full load.

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[7] Power, Sept. 23, 1924, p. 498.

[8] High Efficiency in 1000-Kw. to 6000-Kw. Steam Turbines, by Sanford A. Moss, Power, May 19, 1925, p. 775.

[9] Test of 15,000-Kw. Turbine at Cleveland Municipal Plant, by L. A. Quayle, Power, Oct. 19, 1926.

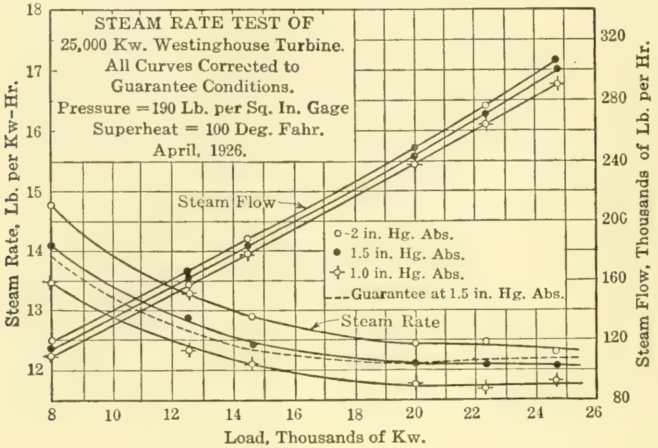

Fig. 329.—Tests of 25,000-Kw. Westinghouse Turbine-Generator at Different Back Pressures.

Incidentally, these values help the beginner to appreciate what it means to have a uniflow engine unit which is of but 1000-kw. capacity give a combined engine efficiency of 72 per cent with only 140 deg. of superheat, such as is shown by curve *B* in Fig. 326.

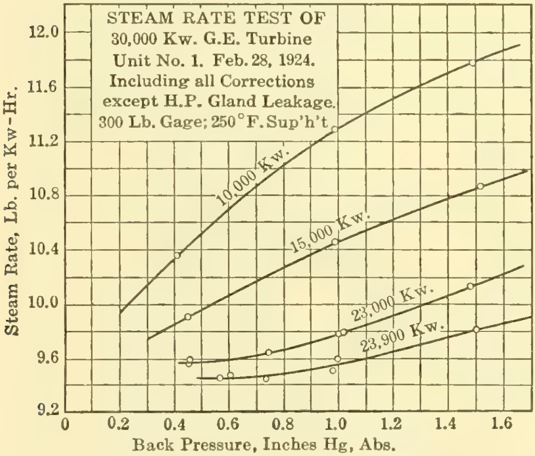

Fig. 330.—Curves Showing Change in Steam Rate with Varying Back Pressure of a 30,000-Kw. G.E. Turbine-Generator.

(**b**) Examples of performance curves of 25,000- and 30,000-kw. turbine-generators are given in Figs. 329 and 330.[10] In both figures the results are given for various loads and condenser pressures. The first unit was designed to have its best thermal economy at a load of 20,000 kw.,

[10] Proc. Nat. El. Lt. Assoc., 1926, p. 1051, and 1927, p. 1188.

above which the overload valve opens. In Fig. 330 the curves show clearly, among other things, that the second turbine was much better able to utilize the very unusually low exhaust pressures, such as 0.4 in. Hg abs., at low load than at full load. This is due to the relatively large leaving loss in the last stage when the unit is under heavy loading, because the volume of the steam then is so great that the exit velocity must be higher than that for which the turbine was intended. This set of curves also shows clearly that the correction for variation in exhaust pressure is not the same for all loads. These examples will suffice to show in general the characteristics of large turbine performance.

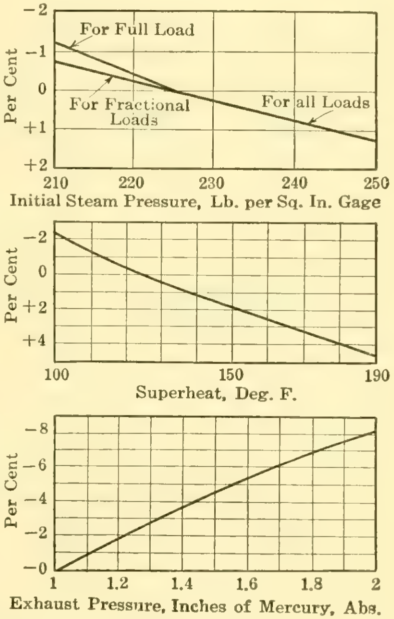

Fig. 331.—Steam-Rate Correction Curves for a 15,000-Kw. Reaction Turbine.

(c) In making tests of prime movers, neither the admission pressure or temperature of the steam, nor the exhaust pressure can be maintained exactly at the desired value during the process of securing data, and this is especially true when the unit must be kept in service while the work is in progress, as is nearly always necessary when large turbines are being tested. Hence, in comparing actual steam rates with the guaranteed ones it is necessary to use **certain steam rate corrections,** the amounts of which are usually agreed upon by the manufacturer and purchaser at the time the machine is sold. The same factors are also used when comparing the performances of two different prime movers. The corrections to be made depend in each case on the design of the machines and the operating conditions. For the 15,000-kw. turbine described in **(a)**, the correction curves are given in Fig. 331.

(d) In case definite information is not available from the turbine manufacturer, corrections given by Eq. (373*a*) following may, if small, be used as fairly representative of average values for tests at full load with throttle pressures from 100 to 400 lb. per sq. in., superheats up to 300 deg. fahr., and condenser pressures from 0.75 to 1.5 in. Hg abs.

Let w_a = actual steam rate under test conditions;

w_{a_c} = actual steam rate corrected to guarantee conditions;

Δp = excess of throttle pressure, in lb. per sq. in., over the guarantee;

ΔD = excess of throttle superheat, in deg. fahr., over the guarantee;

Δc = excess of condenser pressure, in in. Hg, over the guarantee.

Then for the actual turbine

$$w_{a_c} = w_a \div [1 - 0.0004\Delta p - 0.0007\Delta D + 0.06\Delta c]. \quad . \quad (373a)$$

For the same range of pressure and superheats an *ideal turbine* would have its corrected steam rate, w_{i_c}, closely expressed in terms of the ideal steam rate, w_i, for the test conditions, as follows:

$$w_{i_c} = w_i \div [1 - 0.001\Delta p - 0.0006\Delta D + 0.12\Delta c]. \quad . \quad (373b)$$

Note that in both of these equations the sign of any term is changed when the correction to that term involves a *decrease* instead of an excess.

The chief purpose in giving the second equation is to make a comparison of the two and thus bring out the logical basis for Eq. (373*a*). A careful analysis shows that the actual turbine cannot utilize an increased throttle pressure nearly as well as does the ideal. This is because the leakage in the actual machine increases with the pressure, and because the engine efficiency is still further reduced by the larger amount of moisture produced by the greater expansion, and also by the throttling action of the governor. On the other hand, the actual turbine gains slightly more than the ideal by an increase in superheat, because the internal friction losses in the real turbine are greatly reduced thereby. Of course, both the ideal and the actual machines have the same increase in available energy due to the extra superheat and throttle pressure.

The actual turbine at full load is seen to utilize the better vacuum only about half as well as the ideal. This is because the leaving losses in the last stage generally become very large when the turbine is operating at full load and low exhaust pressure. In this connection attention is again called to Fig. 330 because it shows that the turbine there considered is not designed to profit by reducing the back pressure at full load to less than about 0.6 in. Hg abs., whereas at light loads the gain is quite large. Turbines and their condensers should be proportioned with respect to the vacuums that are the most economical to use in the different seasons of the year under their various conditions of loading.

(e) The **effect of size** upon the combined engine efficiency of turbine-generator units operating with comparatively moderate steam pressures and temperatures is clearly shown by the *Curves of Engine Efficiencies* in Figs. 332 and 333. The data from which the first curve was drawn are given in Table XVI, and each point plotted is numbered to agree with that in the table. These figures serve to summarize what has previously been stated regarding the rather poor performances of small turbines relative to those of large ones, and to show what efficiencies have been actually obtained in practice. The difference in the generator efficiencies of the units given in Table XV was probably not more than 5 per cent, as may be seen by referring to Fig. 324. The size of turbine as related to the use of high throttle pressures is discussed in Sect. **275.**

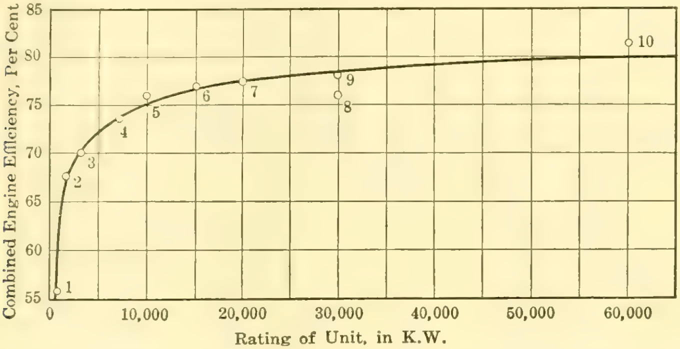

FIG. 332.—Typical Engine Efficiencies of Modern Condensing Turbine-Generators of Various Ratings.

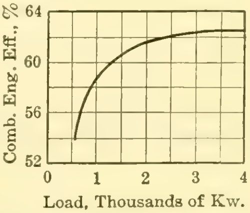

FIG. 333.—Typical Engine Efficiencies of Small Non-condensing Turbine-Generators.

(f) The **rating** of large steam turbines has now become rather well established on the basis of the *maximum load* they will carry, rather than on the basis of maximum thermal economy, which is the more logical and customary one for other heat engines. As a result of this custom the large turbine usually gives its best performance at a load somewhat less than the maximum output. The relation between the most efficient load and the maximum capacity is fixed in the original design, preferably being based on variation in loading expected throughout the year. The full load on large turbines is generally obtained by admitting extra steam (by means of *secondary, tertiary or other inlet valves*) to later stages than the first one. This naturally reduces the engine efficiency some-

TABLE XVI

TYPICAL EFFICIENCIES OF MODERN STEAM TURBINE-GENERATORS OF VARIOUS RATINGS

No.	Rating, Kw.	Average Load		Condition of Steam at Throttle			Exhaust Pressure, in. Hg Absolute	Steam Rate of Unit, Pounds per Kw-Hr.	Engine Efficiency of Unit, Per Cent	Thermal Efficiency of Unit, Per Cent	Energy Consumption of Unit, B.t.u. per Kw-Hr.	Reference
		Kw.	Per Cent Rating	Pressure, Pounds per Square Inch Absolute	Temperature, Degrees Fahr.	Degrees of Superheat						
1	625	631	101	215	487	99	2	16.92	55.8	16.9	20,200	Power, Aug. 26, 1924, p. 319.
2	1,500	1,469	97.8	223	583	192	1.38	12.42	67.8	22.2	15,400	Eng'g, Sept. 10, 1920, p. 345.
3	3,000	3,500	117	240	548	151	1	11.7	70.2	23.4	14,600	Power, Oct. 28, 1924, p. 680.
4	7,000	7,321	105	205	568	184	0.97	11.43	73.6	23.7	14,400	Eng'g, Aug. 22, 1924, p. 254.
5	10,000	8,228	82.3	212	632	245	1.50	11.3	76	25.6	13,300	Eng'g, Nov. 14, 1924, p. 668.
6	15,000	13,853	92.3	272	618	209	1.40	10.5	77	25.6	13,300	Power, Jan. 27, 1925, p. 130.
7	20,000	20,000	100	205	659	275	2	11	77.6	24.3	14,100	Eng'g, Oct. 15, 1926, p. 465.
8	30,000	22,200	74	338	626	198	1.93	10.73	76	25.3	13,500	Mech. Eng'g, April. 1926, p. 366.
9	30,000	24,959	83.2	241	670	272	0.88	9.68	78.4	27	12,600	Trans. A.S.M.E., 1925, p. 1041.
10	60,000	40,332	67.2	219	528	138	1.73	11.02	81.5	25.4	13,400	Trans. A.S.M.E., 1922, p. 377.

what, but it enables the turbine to carry the extra load during short periods when the thermal economy is a minor consideration. From Table XVI it may be noted that all of the large units listed therein have the test data given for loads much less than the maximum.

273. Bleeder and Mixed-Pressure Turbines.—(a) A bleeder turbine with single extraction consists of a high-pressure element large enough to generate by itself the rated power of the unit even when the full amount of heating steam is bled, combined with a low-pressure element which operates at its full capacity only when no bleeding occurs and the power demand is the maximum. The curves for the total steam consumption of the respective high- and low-pressure elements can each be determined independently and then be combined to determine the corresponding curves for the complete unit operating with various amounts of bleeding,[11] due allowance being made for rotational losses of idle wheels and for the steam passed through the low-pressure part for cooling the interior, which would otherwise become overheated by the rotational losses; or, the necessary data can be obtained by testing the complete unit under all the conditions of operation. The performance curves for mixed-pressure turbines and for combined bleeder and mixed-pressure turbines can be obtained by similar methods.

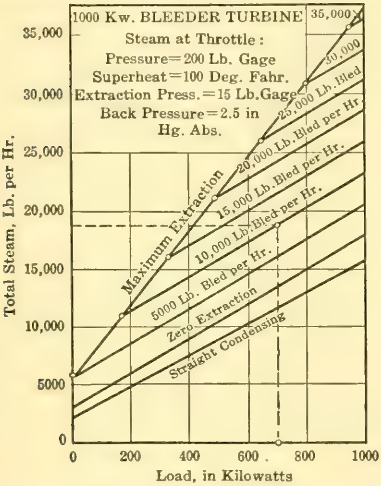

Fig. 334.—Total Steam Consumption of a 1000-Kw. Bleeder Turbine-Generator.

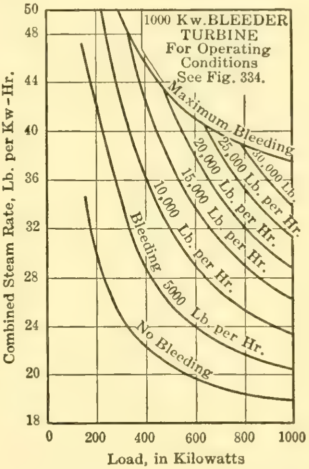

Fig. 335.—Steam Rates Corresponding to Fig. 334.

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[11] Figuring Steam Requirements for a Combined Bleeder-Mixed-Pressure Turbine, by J. Y. Dahlstrand, Power, July 14, 1925, p. 42.

(b) An example of the **total steam consumption curves** of a **bleeder turbine** is given in Fig. 334, which is for a 1000-kw. unit.[12] These consumption curves show that when this turbine operates with zero extraction it uses slightly more steam than when used as a "straight condensing" unit. This means that with the bleeder control valves or grid removed, the operation is slightly better than when they are left in place but are in such position that no steam can be bled. When operated without any power being generated by the low-pressure element, the total steam used per hour is a little larger than the amount bled, as is shown by the points on the maximum extraction line. This is because of the ventilating steam used in the idle part.

The **combined steam rate** of this 1000-kw. bleeder-turbine-generator is given by the curves in Fig. 335, as obtained from Fig. 334.

(c) From curves similar to the foregoing, the **engine efficiency** of such a unit, for any load and any rate of bleeding, may be readily calculated. Thus, let

w = actual combined steam rate of the unit in lb. per kw-hr.;
m = fraction of w that is bled at the pressure p_b;
a = the state of the steam at the throttle;
b and c = the states of the steam resulting from isentropic expansion from a to the bleeder pressure and condenser pressure, respectively.

Then, for the unit having only *one* bleeding point,

$$\text{Comb. eng. eff.} = \frac{3413}{w[h_a - h_c - m(h_b - h_c)]}. \quad . \quad . \quad (374)$$

For the bleeder turbine alone the engine efficiency may be found by dividing the result obtained from Eq. (374) by the generator efficiency, but generally it is the performance of the complete turbine-generator unit that is desired.

By using Eq. (374), and the curves in Fig. 335 the combined engine efficiency of this 1000-kw. unit is found to be about 55 per cent at full load regardless of the amount bled. This may be considered as a very satisfactory performance for a small bleeder turbine but somewhat higher values may generally be expected in larger sizes of units, or those designed for only partial extraction.

(d) An illustrative performance chart for a **mixed-pressure turbine** is given in Fig. 336 for a 500-kw. unit. When the output of the unit is 360 kw. with the low-pressure element receiving 10,000 lb. of by-product steam per hour, the chart shows that 12,500 lb. of steam, per hour, flow to the condenser, that the 10,000 lb. of by-product steam generates 260 kw. in the low-pressure section, and that the 2500 lb. of live steam generates 100 kw. in passing through the whole unit.

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[12] Redrawn from Power, March 30, 1926, p. 477.

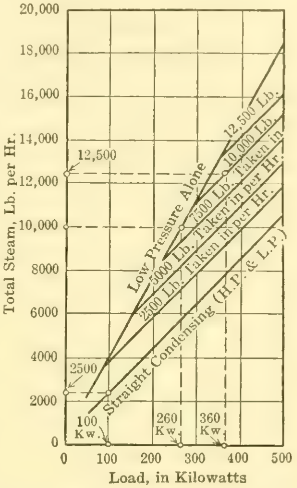

FIG. 336.—Typical Performance Curves of a Mixed-Pressure Turbine.

274. Regenerative Units.—(a) Any steam prime mover from which steam is bled at one or more points for the purpose of heating feedwater, operates on the *Regenerative Engine Cycle* which was shown in Chapter XVIII, Part I, to be thermodynamically far superior to the Rankine, especially at high pressures. This cycle has been extensively used in central station units, and its application becomes more and more important as the steam pressure and size of unit are increased. For this there are two main reasons:

First, the *thermal efficiency of the unit is increased* as the pressure becomes higher, provided the turbine is of sufficiently large capacity. The explanation for this is as follows: The thermal efficiency depends on both the efficiency of the *ideal* cycle and on the engine efficiency of the *actual* machine. With rise in initial pressure, the regenerative *cycle efficiency increases* greatly because this pressure elevation affords better opportunity to bleed more steam for feedwater heating and thus to reduce the amount of waste energy going to the condenser; the limit to the proportion of the total steam that may be bled to the feedwater heaters in the ideal cycle is determined by the boiler pressure, which fixes the capacity of the feedwater to absorb heat without vaporizing. The *engine efficiency* of the actual machine, on the other hand, *decreases* somewhat when the initial pressure is raised appreciably, but the larger the machine the less is this effect. If the turbine is made of great enough capacity, however, the very marked improvement in the cycle efficiency is more than sufficient to overcome the decrease in the engine efficiency, and there results a net thermal gain, with the corresponding reduction in the expenditure for fuel.

Second, the size and cost of the turbine exhaust shell becomes less for a given capacity when the regenerative cycle is substituted for the Rankine, and especially is this so if the steam pressure is also increased. This is because the bleeding of the large proportion of the steam thus made possible decreases the amount flowing through the exhaust shell of the turbine and thereby reduces the necessary size and cost not only of this portion of the machine, but also of the condenser and related parts as well. This saving, especially in large units, usually will be greater than the added expense for the regenerative feedwater heaters, and the extra piping, valves, and pumps required. Further there may be also a reduction in the cost of operating condenser pumps and other apparatus affected.

(b) Incidentally the substitution of the regenerative cycle for the Rankine increases the *maximum power output obtainable with the largest size of turbine exhaust shell* that may be employed with given operating conditions, this being an important factor in very large plants in which units of the greatest capacity obtainable are desired. The length of the last row of blades limits the maximum flow of steam through them with a given back pressure, because the greatest length that these blades can safely have with given rotative and blade speeds is determined by their structural strength. Hence there are definite limits as to the exhaust area, or size of exhaust shell, that can be used with prescribed conditions. If no steam is bled for any purpose, the throttle flow must be kept down to the maximum value that can be handled by the last blades; therefore the greater the amount of steam bled from the various stages of the turbine after having done work therein, the larger is the maximum power output with a given exhaust area. Hence the regenerative cycle and high throttle pressures afford considerable advantage regarding large capacity over the Rankine cycle and low initial pressure.[13]

(c) The **losses involved in the regenerative cycle** will now be briefly considered. Each unit of mechanical energy delivered by a turbine from steam that is condensed in a regenerative heater, is generally produced with a very *high thermal efficiency*, because none of the energy of this steam is thrown away to the exhaust. This efficiency, however, is not 100 per cent, because the following losses involved with each portion of the steam bled cannot be made zero:

1. The *thermodynamic losses* due to using but relatively few heaters instead of an infinite number, and due to the drop in pressure between turbine and heaters and to the difference in temperature between steam and water in surface heaters.

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[13] The application of very high pressures with regeneration, or otherwise, may require the use of inter-stage reheating, or high steam temperatures, or both. The discussion of the special factors involved will be given consideration in Sect. **276**.

2. The *internal losses of the turbine* due to leakage, turbulence, and friction of the steam in passing partially through the turbine, i.e., up to the point of extraction.
3. The *losses in the steam and feedwater circuits* due to fluid friction, turbulence, and leakage.
4. The *mechanical losses* in the turbine and feed pumps.
5. The *heat dissipation losses* due to the transfer of heat from the steam lines, turbine casing, feed lines, heaters, and pumps.

The losses in the boiler room, in the generator, and in all auxiliaries are also involved when considering the output of the generator per unit of fuel burned.

(d) The **number of heaters used** and *the feedwater temperature attained* both have an important bearing on the thermal performance of the turbine and station. The complication and expense generally keep the number of heaters down to 2 or 3 for medium-size units, whereas 4 or sometimes 5 may be profitably installed with large units operating under high pressure steam.

For the ideal cycle using an infinite number of heaters the higher the feedwater temperature the better is the cycle efficiency. Consequently, at first thought, one might believe that in the actual case, the higher the feedwater temperature the greater will be the plant thermal efficiency; but one must remember that the station is operated for the sole purpose of delivering electrical or mechanical energy derived from fuel by means of imperfect apparatus involving the losses enumerated in the preceding section. It is possible, therefore, in an actual plant that the high-pressure heater may receive steam bled at too high a pressure, thus resulting in a very high feedwater temperature, but obtained at the expense of considerable reduction in the work done by each pound of fluid that completes its cycle. On the other hand, the high-pressure heater may receive steam at such a low pressure that the thermal economy of the station will be less than it should be for the apparatus installed.

(e) The **best feedwater temperature** to use and the number of heaters to install are matters that require considerable study. In general, each heater added in the system improves the performance, but in less amount than does the preceding one, whereas it costs just as much; hence, according to the law of diminishing returns, there is, in each case, an economic limit to the number of heaters that should be used. The relation between this number and the feed temperature, the throttle pressure, the throttle temperature, the exhaust pressure, the engine efficiency, and the energy consumption of a turbine is rather involved and cannot be presented in simple form. However, the results that may be expected from quite efficient modern turbine-generator units of large size operating with different numbers of heaters may be rather easily obtained by the aid of the curves given in Figs. 337, 338 and 339, as calculated by Dr. A. De Smaele.[14]

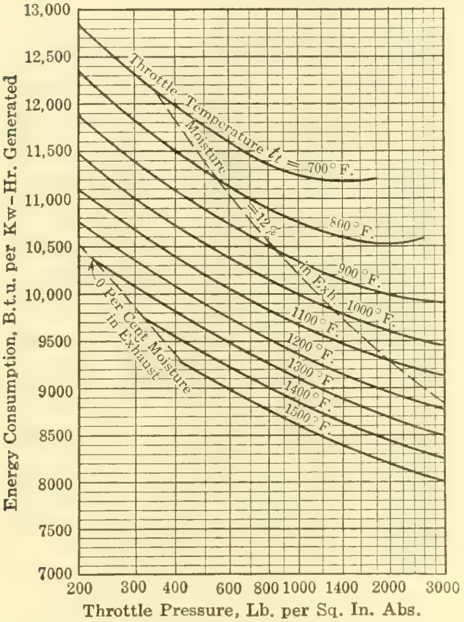

Fig. 337.—Calculated Performance Curves of Large Turbine-Generators Without Bleeding or Reheating. (Leaving loss = 10 B.t.u./lb.; loss in pressure due to throttling = 5 per cent.)

The curves in Fig. 337 show the computed energy consumption, in B.t.u. per kw-hr., of large turbine-generators when operating without bleeder heaters, the units having an assumed mechanical-electrical efficiency of 94 per cent, stage efficiencies of 84.5 per cent with superheated steam and $84.5x$ with wet steam in any stage having the average quality x, an exhaust pressure of 1 in. Hg abs., throttle pressures as shown by the abscissas, and temperature as given on the curves. If similar turbines could be operated with an infinite number of perfect heaters that, without losses of any kind, bring the feed temperature to the saturation temperature corresponding to throttle pressure, the per cent reduction in their energy consumptions due to such heaters would be as given by the curves in Fig. 338. For example, with steam superheated to 800 deg. fahr., these curves show that the energy consumption of the turbine would be reduced about 10 per cent at 200 lb., or over 20 per cent at 3000 lb. by perfect regenerative feed heating. However, such a system cannot be applied, because only a finite number of heaters can be used, and each one involves a definite loss due to drops in pressure and temperature, and adds to the cost of the unit. Consequently the reduction in energy consumption in any actual case will be only a certain portion of that shown by the curves in Fig. 338. For the actual heaters a pressure drop of 5 per cent between the turbine and heater, and a terminal temperature difference of 7 deg. fahr. may reasonably be assumed; and on that basis the curves in Fig. 339 may be used to find the reduction in energy consumption, due to either 1, 2, 3, or 4 heaters, as a percentage of that obtainable with perfect regeneration, as given in Fig. 338.

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[14] Thesis, "The Utilization of High Pressure and High Temperature Steam for the Production of Power," Cornell University, June, 1930.

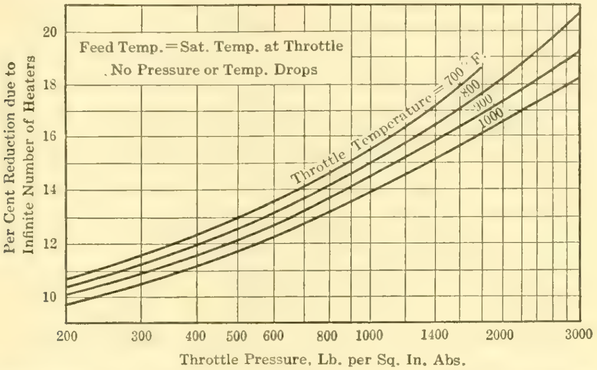

FIG. 338.—Reduction in Turbine Energy Consumption Due to Applying an Infinite Number of Perfect Heaters. Per Cent of Consumption Given in Fig. 337.

The curves in Fig. 339 also show that for any given number of heaters there is one feedwater temperature that gives the best thermal results, but that this point is not sharply defined and hence there is considerable variation possible in this temperature without seriously affecting the economy of operation of the turbine. They also show very clearly how much more important is one heater than none, and that each additional heater is of less value than the previous one. By basing the curves in Fig. 339 on the percentage of the maximum possible rise in feedwater temperature they become

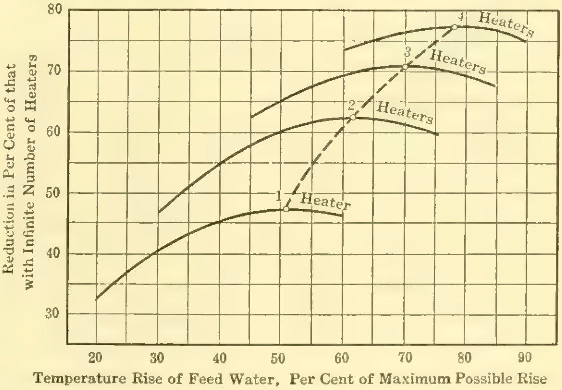

FIG. 339.—Energy Reduction with Finite Number of Heaters, in Percentage of Values Obtained from Fig. 338.[15]

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[15] Based on the following assumptions: Pressure drop from extraction nozzle to heater = 5 per cent; terminal temperature difference in heater = 7 deg. fahr. The maximum possible temperature rise is equal to the saturation temperature at throttle pressure minus the temperature of the condensate in the hotwell.

applicable with any hotwell pressures. They are valid for throttle pressures from 200 to 3000 lb. per sq. in., and for any throttle temperature from 700 to 1000 deg. fahr.

As an example of the use of the curves in Figs. 337, 338, and 339, suppose one desires to estimate the best performance and feedwater temperature of a large turbine operating with four heaters, a steam pressure of 400 lb. per sq. in. abs., a temperature of 800 deg. fahr., and an exhaust pressure of 1 in. Hg abs. The procedure would be thus: By the aid of a steam table, sat. temp. at 400 lb. − sat. temp. at 1 in. Hg = 445 − 79 = 366 deg. fahr. Then from Fig. 339, for 4 heaters, the minimum energy consumption of the turbine would be obtained with a rise in feedwater temperature of about 78 per cent of 366, or 285 deg. fahr. The feedwater would then leave the last heater at 285 + 79, or 364 deg. fahr. With the steam in this heater at a saturation temperature about 7 deg. higher than that of the water, its temperature would be 371 deg., which means that the steam pressure in the heater is 176 lb. per sq. in. abs. With a 5 per cent drop in pressure between turbine and heater, the bleeding from the turbine would therefore be at a pressure of about 185 lb. By reading both Figs. 338 and 339 the energy consumption of the turbine-generator is found to be 0.12 × 0.78, or 9.3 per cent, less than if no heaters had been used; therefore, by the aid of Fig. 337, the minimum energy consumption that could be expected from this unit with 4 heaters would be (1 − 0.093) × 11,500, or about 10,430 B.t.u. per kw-hr. Similar estimates may be readily made for other conditions.

Special attention is now called to the fact that the preceding discussion has been *dealing only with the turbine unit, including the heaters and generators.* The effect of feedwater temperature, number of heaters and steam pressure on the heat rate per kilowatt-hour of the *whole* plant, or *station heat rate,* will be considered in Sect. **279** (**c**) after showing some results of regenerative turbine units which were tested separately.

(**f**) **Tests of turbine units** using the regenerative cycle have until recently been commonly made with the feedwater heaters not in use, i.e., the tests have been performed with the unit operating "straight condensing." The reason for this has been the thought that there is greater difficulty encountered in conducting the test and in analyzing the results when the heaters are in use. But if the manufacturer has made his guarantees on the non-regenerative conditions, and has designed the unit to give its best results on that basis, the machine does not have the proper proportions for giving the best thermal economy when the heaters are in use. Methods of testing and interpreting the results of regenerative performances are now better understood; hence future tests should be performed on that basis. The operating force of a station is glad to have the unit tested with the feed-heating system in use because the normal operation of the station is not so seriously affected during the test. In fact, the saving in fuel by making the test on the regenerative basis will be an item of considerable importance with a large unit if the test is a reasonably long one.

(g) **Data from actual regenerative tests** for three different sizes and makes of turbines are shown by the curves in Figs. 340, 341, and 342,[16] in each of which are given the condition of the steam at the throttle, the exhaust pressure, the number of heaters, and the rating of the unit represented. In addition to the main curves others are also given to show how the steam pressure in the high-pressure heater, the amount of steam bled, and the rise in feedwater temperature all become greater as

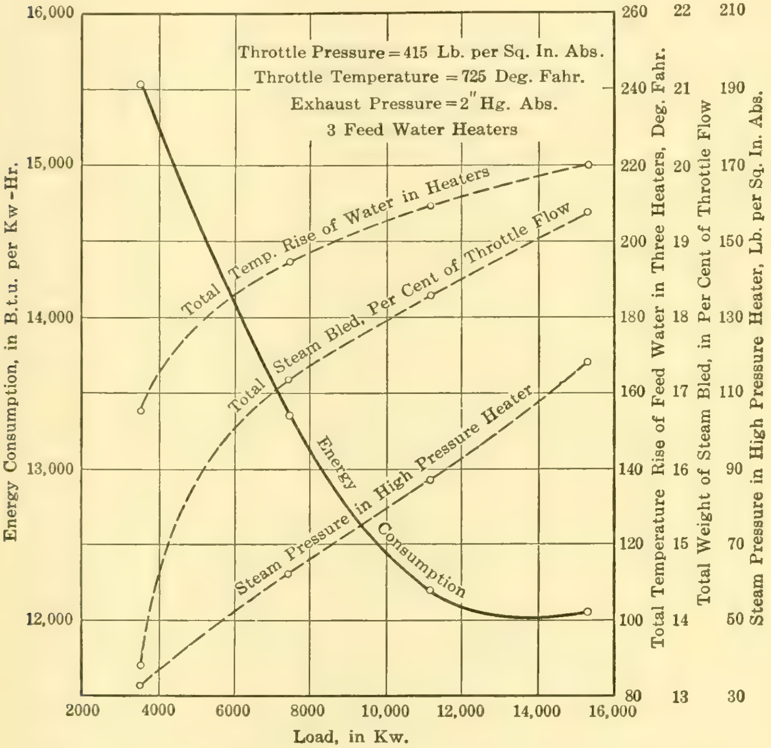

FIG. 340.—Test of a 15,000-Kw. Allis-Chalmers Regenerative Unit.

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[16] Data obtained from the National Electric Light Association Proceedings, 1929, Vol. 86, p. 1370.

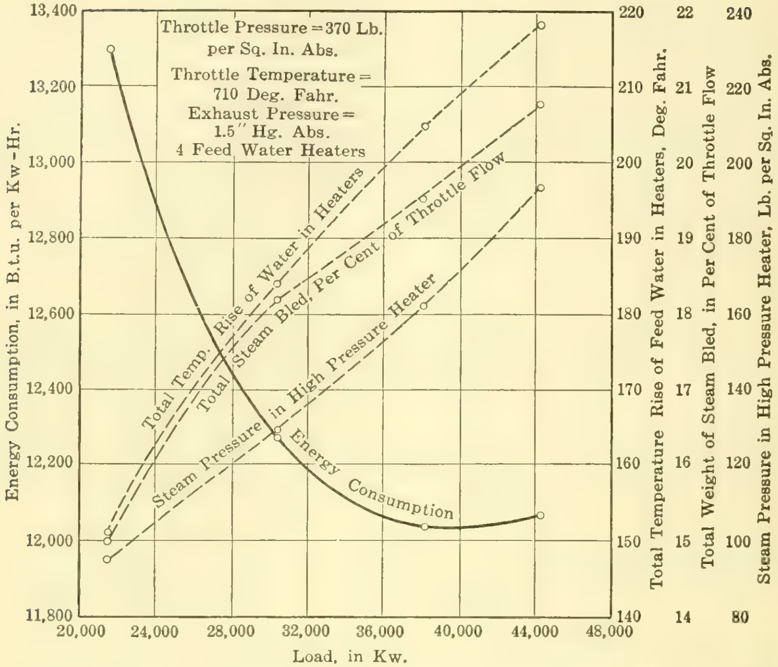

FIG. 341.—Test of a 47,250-Kw. Westinghouse Regenerative Unit with Single Over-Load Valve.

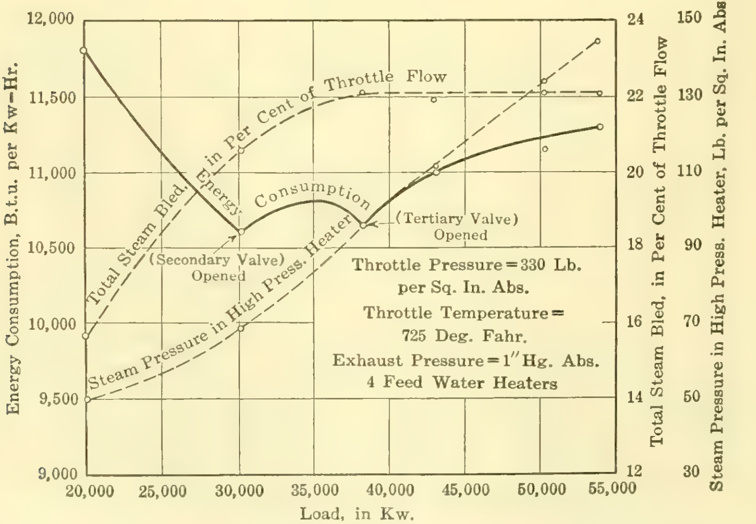

FIG. 342.—Test of a 50,000-Kw. General Electric Regenerative Unit with Secondary and Tertiary Valves.

the load on the turbine is increased. For the given conditions the curves in these figures show very satisfactory performances, but they do not represent the best results that are likely to be attained with further development in the next few years. The higher steam pressures and temperature in prospect, combined with larger size units, will improve the energy consumption very materially, as will be shown in Sect. **275.** In this connection attention is called to the fact that turbine manufacturers do not offer the *best* turbines they know how to make, but the best they can build for a certain price. In other words, all large turbines are sold on a competitive basis in which the *cost* of the unit is very carefully considered in connection with its energy consumption curve over a considerable range of load.

275. Application of High Pressures and Temperatures to Turbines.—(**a**) The previous sections of this chapter have shown a number of *actual* performance curves that were obtained by tests, but this one will deal largely with *calculated* results, because in it will be discussed, in many cases, applications of higher pressures and temperatures than have yet been used or for which experimentally determined data are as yet not available. In this connection, one should remember that all engineering progress is made by developing new ideas, and that no modern manufacturer would consider seriously the building of a new design of turbine, or other machine, unless he had confidence in the results of the engineering calculations that show with reasonable accuracy the performance that may be expected from such a machine. In this respect, steam turbine development to date affords a most gratifying background because year after year the turbine has become larger and better, sometimes so rapidly as to be startling; and the calculated and actual performances have generally been in remarkably close agreement and they may be expected to continue so in the further developments.

(**b**) The use of **high-pressure** steam in large turbines has grown very rapidly during the last few years, mainly because of the fuel saving effected by the greater amount of energy thus made available per unit weight of steam flowing through the machine; but higher pressure accentuates the need for satisfactory methods of preventing or minimizing the serious energy loss and blade erosion troubles due to the **excessive moisture** in the low-pressure stages of such machines. The relation existing between the condition of the steam at the throttle and the moisture in a turbine is shown in Fig. 343 by the broken heavy lines, which are based on the assumptions that the turbine operates with a 5% drop in pressure due to throttling, that no reheater is used, and that the stage efficiency of the turbine is 0.845 with superheated steam and $0.845x$ with wet steam in any stage having the average quality x. For these conditions, this set of curves shows very clearly the need for high throttle temperatures when using even moderately high throttle pressure and low exhaust pressure, if the moisture in the turbine is not to exceed 10 or 12 per cent, which is generally considered as about the maximum allowable for satisfactory operation.

The curves may also be used to show how, without exceeding an allowable value of the initial temperature, the moisture may be kept down to the allowable maximum by reheating the steam one or more times when very high pressures are used. For example, with a throttle pressure of 2000 lb. per sq. in. and a temperature of 850 deg. fahr., expansion to 0.5 lb. back pressure without reheating would result in 18 per cent moisture at the end; but if, upon reaching a pressure of about 300 lb., the steam is reheated at constant pressure to the throttle temperature, and is then expanded to the same terminal pressure, the exhaust will contain only about 8 per cent of moisture. Again, for the same throttle pressure but a lower temperature, the resulting terminal moisture would be still greater; but reheating the steam more than once during the expansion would prevent excessive moisture. But multiple reheating by the usual methods involves considerable extra complication and increased first cost, and cannot be justified unless the yearly load factor, or average load, is very high.

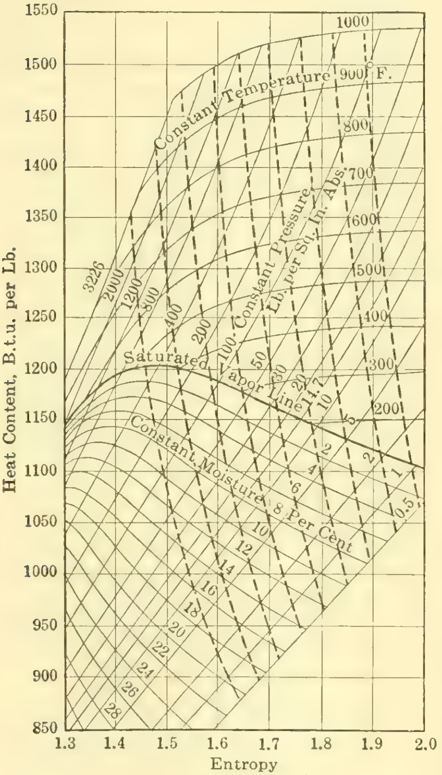

FIG. 343.—Illustrating Condition Curves for Large Steam Turbines.

Any decrease of entrained moisture that can be accomplished by *mechanical* means reduces, of course, the amount of reheating necessary; hence some manufacturers provide within their turbines special devices or arrangements whereby at least a portion of the moisture is separated and drained off; but a high degree of elimination by such means, although particularly desirable, is very difficult to obtain, and the main reliance must therefore still be placed on reheaters. The amount of reheating necessary can also be minimized effectively and simply, within certain limits, merely by increasing the initial temperature of the steam entering the throttle at the given pressure.

(c) Although *high steam temperature* is desirable, both for increasing the amount of energy made available by the ideal expansion from the given initial pressure to the final one within a turbine, and also for reducing the amount of moisture in the lower stages of the turbine, there are certain values beyond which the temperature should not be elevated in practice. The **maximum temperature** that may be safely used in a steam plant is limited by the materials available for making superheater tubes, steam pipes, valves, turbine blades, and other parts affected. Up to the present time (1930) the throttle temperatures in use have seldom exceeded 800 deg. fahr.; and in the more moderate practice it has been from 600 to 750 deg. However, experience has now shown that, when conditions make it desirable, a temperature of 850 deg. fahr. can be successfully used with the customary materials, in the superheater, and some stations with special materials therein have operated for considerable portions of a year with 930 deg. fahr. Experimental work with special alloys indicates that temperatures as high as 1100 deg. fahr. may eventually be applicable when plant and economic conditions are suitable. In general, regardless of the throttle pressure used, the initial steam temperature should be made as high as the construction difficulties, operating conditions, and economic factors justify.

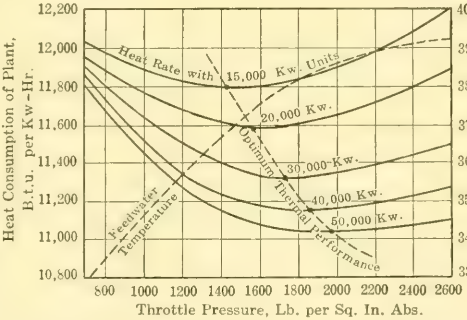

FIG. 344.—Curves Illustrating the Effect of Turbine Size and Steam Pressure on Plant Performance under Certain Conditions.

(d) The **proper steam pressure** to use in a turbine cannot be specified in any simple manner because the economic and thermal considerations involved are dependent also on such additional factors as the steam temperature, method of reheating, load factor, size of the turbine and cost considerations. To show the relation between *steam pressure*, *size of turbine*, and *thermal economy*, the curves in Fig. 344 have been drawn from the data given in the Brown-Boveri Review, Jan., 1930, for a certain set of conditions and special arrangement of turbine.[17] Clearly, the curves of heat consumption have a well defined minimum value for each size of unit and show definitely that very large turbines are needed to give the best thermal performance when extremely high pressures are used. For example, these curves show that, with the kind of turbine considered and with a throttle temperature of 842 deg. fahr., a station with 15,000-kw. units would give its best thermal performance when using a steam pressure of about 1450 lb. per sq. in., but with 50,000-kw. units the optimum pressure is about 2000 lb. For other throttle or reheat temperatures and other turbine types and arrangements the curves would be somewhat similar to those given in this figure, but the points of minimum consumption would be shifted.

The improvement in thermal efficiency with increase in *size of units* is chiefly due to the relative reduction in the percentage of leakage and rotational losses in the high-pressure stages, but the other turbine losses are also lowered somewhat, and furthermore the generator efficiency is also a little better, as already shown in Sect. **270.** When dealing with *very* high pressures the consideration of capacity alone, however, is altogether insufficient to give an idea of the internal efficiency to be expected of the turbine. The *rate of throttle flow per unit of throttle pressure* gives a much better indication; for, although the leakage, rotational and other losses increase with the pressure, the *relative* value of these losses and the turbine efficiency can be kept constant or may even be reduced by making the steam flow sufficiently large. With very limited flow, the reciprocating steam engine may successfully displace the high-pressure steam turbine under certain conditions.

The larger units also have the economic advantage of lower cost and smaller space occupied per unit of power, but the station with several medium-size units is able to handle wide fluctuations in load better than it could with the same installed capacity in fewer but very much larger units. An extremely large unit has the further disadvantage of causing a very serious reduction in plant capacity if it is compelled to shut down for any reason when the station is fully loaded. Thus the proper selection of the size of turbine involves much more than the consideration of the thermal efficiency. This subject will be discussed more fully in a later chapter.

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[17] The conditions assumed in making these computations are: Boiler efficiency, 84 per cent; steam temp., 450 deg. cent. or 842 deg. fahr.; condenser pressure, 0.04 Kg. per cm.^{2} or 0.57 lb. per sq. in. abs.; reheating-regenerative cycle with feed temperature as shown; Brown-Boveri four-cylinder turbines with water drainage; all plant auxiliaries included in the computations.

(e) Although the use of high throttle temperatures is much the preferred method of applying heat for reducing the moisture in the last stages of the turbine if materials are available to stand them, yet, because of such limitation, recourse must be had to other methods when the pressures used are high. Hence the rapidly increasing use in the high-pressure plants of reheating the steam, which will be discussed in the next section.

276. Application of Gas or Live-Steam Reheat.—**(a)** From the early days of their use, multi-cylinder engines have frequently been provided with reheating receivers between cylinders, and considerable test data relating to such machines are available. More recently large turbines using throttle pressures above 450 lb. per sq. in., and temperatures of 750 deg. fahr. and below, have been equipped with reheaters connected between their high- and low-pressure elements. These units also always use regenerative feedwater heating, hence they are of the *reheating-regenerative* type. With the growing application of higher pressures there is an increasing demand for large machines of this type.

The improvement due to adding a reheater to a regenerative turbine lies mainly in the gain due to using therewith the higher throttle pressures without causing a serious reduction in the stage efficiencies and without increasing the erosion in the last stages.

Inter-stage reheating with turbines, or **reheating,** as it is more briefly termed, generally has been accomplished by having the partly expanded steam absorb additional energy either from the boiler gases or from high temperature steam (live steam) taken directly from the main. These two methods are commonly referred to, respectively, as *gas reheating* and *live-steam reheating.* Gas reheating is sometimes also called boiler reheating.

(b) With **gas reheating,** the steam from the high-pressure element is generally piped back to a reheat boiler where it is usually reheated to the same temperature as that at the throttle, and with this assumption the question arises as to what is the **best reheat pressure,** or pressure at which the steam should be withdrawn from the turbine for the reheating. For *one gas reheating to throttle temperature, the best thermal efficiency of the turbine is closely obtained by reheating at the pressure attained when the temperature of the steam has become about one-half of the throttle temperature,* provided, however, that the steam at the beginning of reheating is still superheated about 10 deg. fahr.[18] This simple general rule is the result of very careful calculations made by Dr. De Smaele and is believed to be a safe one on which to make further calculations that involve the economics of the installation. Usually these economic considerations will show that *it is desirable to reheat the steam at a pressure somewhat higher than that giving the best thermal efficiency,* because the first cost of the reheating system is thereby reduced. In this connection it should be stated that, considering all things, boiler reheating at pressures much less than 400 lb. per sq. in. is now probably seldom justified economically.

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[18] Wet steam should not be allowed to enter a boiler reheater because scale is likely to be deposited therein from the moisture, and because the water might even pass entirely through the reheater, thus causing damage to the turbine, piping or the reheater itself by reason of the high velocity.

Gas reheating requires the use of long pipes of large diameter leading from the turbine to the boiler and back again. This involves problems not only of cost and large pressure drop (in some cases as much as 10 per cent) but also those connected with the provision for the expansion and contraction of the piping, and with the storage within the system of a large quantity of steam which must be kept out of the low-pressure element in case of a sudden dropping of the load. When this method is used, the steam is generally reheated but once, because of the expense of equipment and complications involved.

(c) With **live-steam reheating** the equipment in the plant, especially the piping, is greatly simplified, but the thermal efficiency is somewhat less than with gas reheating. Since the live steam reheater may easily be placed near the turbine the use of very low reheating pressures, which involves handling large volumes of steam, and the adoption of more than one reheating, may be justified. Wet steam may also be reheated without difficulties with this system of reheating.

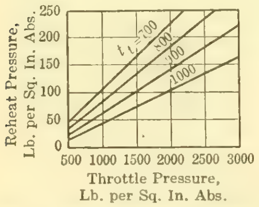
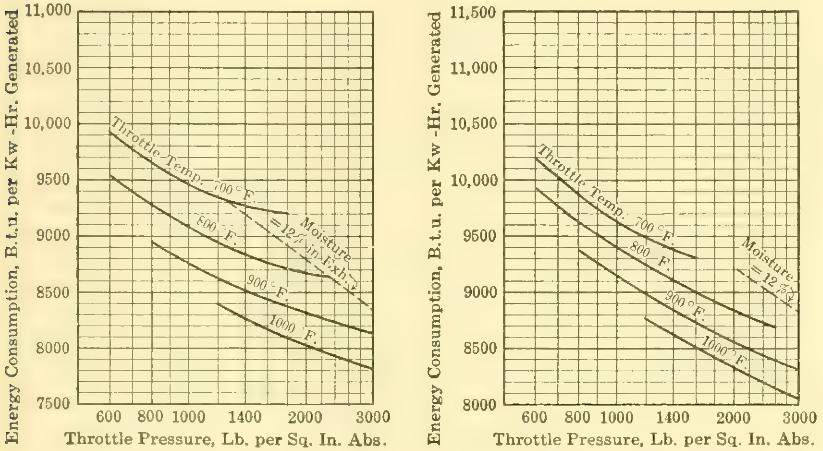


Fig. 345.—Reheat Pressures Giving Maximum Thermal Efficiency of Turbine with One Live-Steam Reheat.

To secure the highest thermal efficiency of the turbine having one live-steam reheater, the **proper reheater pressure** to use for any given throttle condition may be found from De Smaele's curves given in Fig. 345, which were based on the assumptions that the steam is reheated to the saturation temperature corresponding to the throttle pressure and that the reheated steam suffers a 3 per cent drop in pressure.

(d) The **results obtainable** with one *live-steam* reheating and one *gas* reheating can be compared by means of the curves (*a*) and (*b*) in Fig. 346, which are based (1) on the same stage efficiencies that were used in determining the condition curves in Fig. 343, (2) on use of the reheating-regenerative cycle, and (3) on the gas reheating pressure taken as given by the rule on page 117, and the *live-steam* reheating pressure as given by the curves in Fig. 345. From the two sets of curves, the energy consumption of the turbine-generator over a wide range of pressures and temperatures is seen to be from 2 to 4 per cent better with gas reheat than with live-steam reheat.

In both figures the 12 per cent exhaust moisture line is drawn to indicate the region above which the throttle pressure should not be carried for any given throttle temperature. For example, Fig. 346 shows that with a throttle temperature of only 700 deg. fahr. and one gas reheat the maximum pressure that should be used is a little under 1300 lb. if 12 per cent is the moisture limit adopted; but if a throttle temperature of 1000 deg. can be used, then either system of reheating will permit the throttle pressure to be carried above 3000 lb. without exceeding this limit.



(*a*) One Gas Reheat.
Reheat Temp. = Throttle Temp.

(*b*) One Live-Steam Reheat.
Reheat Temp. = Sat. Temp. at Throttle.

Fig. 346.—Estimated Energy Consumption of Turbine-Generators Using the Reheating-Regenerative Cycle with Four Feedwater Heaters. (Exhaust Pressure = 1-In. Hg Abs.; Electro-Mechanical Eff. = 0.94.)

A *combined system* of reheating is sometimes used in which the steam to be reheated is first passed through a live-steam reheater and then through a gas reheater; the supply of live steam to the first heater is thermostatically controlled so as to maintain constant final temperature of the steam, even with variable gas temperature and steam flow.

Using two or more steam reheats lowers still further the moisture resulting with given initial pressure, or makes a higher throttle pressure applicable with a fixed moisture limit; but using *live* steam to reheat *very* low pressure steam is not thermodynamically desirable, and is probably seldom justified economically. When two or more reheats are adopted, however, it may be desirable to use a combination of heating by flue gas in the first heater and by means of steam in the second or succeeding ones. Using steam for reheating makes it possible to locate the reheater near the turbine, the length of low-pressure piping then being negligible, but the condensation from this heater should be returned to the boiler, and this requires a special pump designed for a relatively small volume of water and a high pressure and temperature. Some engineers favor placing the live-steam reheater directly over the boiler so that the condensate from the heater may drain into the boiler, thus avoiding the use of a pump.

278. Methods of Expressing and Determining the Performances of Steam Stations.—(**a**) All steam power plants, whether large or small, are built for the purpose of transforming into mechanical or electrical energy some of the heat energy derived from the burning of fuel. Consequently the thermal performance of such a *station* is measured by the amount of energy delivered, or sent out by the plant, from each unit of energy supplied by the fuel. Sometimes the station performance is expressed by the energy output *per unit weight of fuel consumed*, and this affords a most satisfactory measure of performance for the owner of the station provided he is concerned only with *one station* and *one fuel*. But engineers are constantly in need of information regarding the relative performances of different stations operating with various kinds of fuels that have widely varying heating values, and for each case the station output needs to be based on the *energy available* from the combustion of a certain weight of the fuel, rather than on the mere *weight* of the latter. Therefore the preferred and more usual method of expressing the station performance is by giving either the **station heat rate,** say in B.t.u. per kw-hr., or the **plant thermal efficiency,** which would be 3413/(wt. of fuel used per net kw-hr. × heating value of fuel).

(**b**) The *station heat rate* depends primarily upon the efficiency obtained from the boiler (including the furnace and fuel burner) and the thermal efficiency of the turbine or engine; and secondarily upon the performance of all of the auxiliaries, such as pumps, fans, and equipment for fuel handling and preparation. The power required to operate the auxiliaries depends considerably upon the losses connected with the transmission of steam, water, flue gases, and air between various parts of the plant. The relative magnitudes of these losses vary considerably with the load and are seldom the same in different plants. Some stations have little time in which to change from carrying a light load to meeting nearly a full one, or even an overload. Such stations must therefore have a large reserve of equipment in standby operation or idle, but ready to start at short notice. This involves losses that are spoken of as *standby losses.*

All of the losses incident to boiler and furnace operation will be treated in considerable detail in later chapters, but for the present discussion the boiler room efficiency may be assumed as some reasonable numerical value, or a general symbol may be used for it; thus the effect of high steam pressure and temperature on the *station heat rate* may be advantageously studied and compared with the *energy consumption* of the *turbines.*

(c) The **energy consumption of a boiler feed pump** is more directly affected by high pressure than that of any other auxiliary in the plant. The importance of the feed pump, combined with its special relation to the energy supplied to the working substance render somewhat questionable the former custom of calling it "an auxiliary"; but, regardless of such doubtful classification, its relation to plant performance needs to be carefully studied.

The boiler feed pump in a central station is usually in series with the hotwell pump, and thus receives water at a pressure that is generally above atmospheric. The *hotwell pump* is specially designed to receive the water at the very low pressure and the correspondingly low boiling point in the hotwell of the condenser, and to pump against a low head without having serious trouble due to cavitation caused by the formation of vapor. The *feed pump,* on the other hand, is designed primarily to deliver water at a pressure appreciably greater than that in the boiler, because there are losses in the pipe line. Both the hotwell and feed pumps, unless quite small, are generally of the centrifugal type, and are driven by small steam turbines or electric motors. If both pumps are electrically driven, then the *energy input to their motors* necessary **for each kilowatt-hour produced by the main unit** would be, in kw-hr.,

$$E_p = \frac{144 w \Delta p \overline{V}}{778 \times 3413 e_m e_p} = \frac{w \Delta p \overline{V}}{18{,}450 e_m e_p}, \quad . \quad . \quad . \quad (375)$$

where w = weight of water handled by the pump per kw-hr. of the main unit;

Δp = rise in pressure through pumps, in lb. per sq. in.;

$\overline{V}$ = specific volume of the water leaving the pump;

e_m = efficiency of the motor;

e_p = efficiency [19] of the pump.

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[19] This term is here taken in accordance with the pump manufacturers' general usage that does not credit the energy delivered by the pump to the water in the form of heat. Strictly speaking, with boiler feed pumps this energy should be taken into account, certainly so if the pressures pumped against are high.

The value of w is usually taken the same as the steam rate of the turbine, but the method of handling the make-up water, the drains from live-steam reheaters and feedwater heaters will affect this value slightly. The value of Δp is equal to the difference in pressure between the boiler and hotwell plus the loss due to friction and throttling in the entire feed line system. For a turbine exhausting into a condenser, the throttle pressure, p_t, which is always appreciably lower than the boiler pressure, may very conveniently be chosen as the basis on which to estimate Δp. Thus, a reasonable assumption would be that $\Delta p = 1.3$ or 1.4 times p_t. If the latter value be chosen and the combined efficiency of motors and pumps be assumed as 50 per cent, Eq. (375) becomes, in the same units as before,

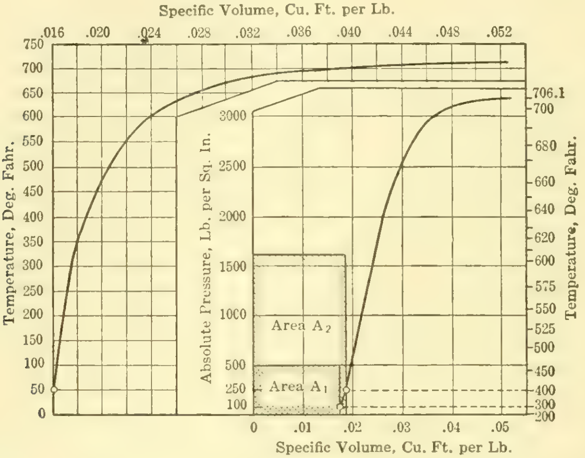

FIG. 348.—Pressure, Volume, and Temperature of Saturated Water.

$$E_p = \frac{wp_t\,\overline{V}}{6600} = 152\,(10)^{-6}\,w\,p_t\overline{V}. \quad . \quad . \quad . \quad . \qquad (376)$$

Note, from this equation, that if the steam rate, w, of the main unit and the throttle pressure, p_t, are both very high, the energy required by the pumps also becomes quite considerable, even though the specific volume of the water in the pumps is always a small quantity. For all pressures that are likely to be used in steam boilers, the specific volume of water, whether saturated or not, depends almost entirely upon the water temperature. The relation of the volume and temperature of saturated water is shown by the upper curve in Fig. 348. Note the rapid increase in volume at high temperature. For example, comparing the volume at 100 deg. fahr. with those at 300, 500, and 700 deg., the curve shows increments of about 8, 27 and 145 per cent, respectively. The relations between the pressure and specific volume of saturated water are represented by the lower curve in Fig. 348; and a scale of saturation temperatures has also been added, but the latter values at low pressures cannot be read from it.

(d) The **energy recovered by the feedwater** from that delivered to the feed pump motor should now be considered because of its importance at high pressures. Since the lower curve in Fig. 348 is drawn to scale on pressure-volume coordinates, the work done upon a pound of feedwater in passing through the pumps may be easily represented for any specified pressure and temperature, by areas on the figure, if it is assumed, as one reasonably may do in most cases, that the compressibility of the water and the suction pressure to the hotwell pump are negligibly small. Thus, for example, the area A_1 in Fig. 348 represents this work for a delivery pressure of 500 lb. and a temperature of 300 deg., while the area A_2 shows the corresponding value for 1600 lb. and 400 deg. Note that, for these areas, the pressures and temperatures should be measured at the delivery of the pump because centrifugal pumps may produce much turbulence, thus increasing both the temperature and specific volume of the water.

Such areas as A_1 and A_2 do not represent the energy required by the pump motors, i.e., E_p of Eq. (376), because all of the motor losses and those of the pump bearings and part of those of the stuffing boxes, together with a small amount of heat from the pump casing, are dissipated to the atmosphere or other cooling medium. These losses constitute the difference between the energy supplied to the pump motors, E_p, and that returned to the feedwater, E_w, by the pumps. The ratio of E_w to E_p, which may be called the *recovery ratio*, r, depends upon the size, speed, load, pump pressure, and the design of pump and motor, so that no single numerical value can be expected to fit all cases accurately; hence, for any specific case, especially one involving high pressures, this information should be determined. In general, then, from r and Eq. (375) the *energy absorbed by the feedwater* from the pumps, for each kilowatt-hour output of the main unit, is, in kw-hr.,

$$E_w = r\,E_p = \frac{rw\,\Delta p\,\overline{V}}{18{,}450 e_m\,e_p}. \qquad (377a)$$

If r is taken as 50 per cent, which is certainly a very conservative figure for most cases, as it might be much higher, and with the other values assumed as in Eq. (376), then, with the same units,

$$E_w = 76\,(10)^{-6}\,wp_t\overline{V}. \qquad (377b)$$

Since each of the Eqs. (375), (376), and (377) gives the fraction of a kw-hr. of energy involved with *one* kilowatt-hour output of the main turbine-generator unit, each also gives a value that represents its corresponding *fraction* of the generator output at the load for which w is the actual steam rate; and this is the most convenient way to use these results.

Strictly speaking, the net rate of energy consumption of the turbine and the feed pumps combined is greater than that of the turbine alone by the amount $E_p - E_w$, instead of by E_p. The true credit of all the energy returned to the boiler room by all feed line pumps, as well as by the regenerative heaters, can be accurately based on the heat content of the non-saturated water[20] leaving the last feed pump, assuming it to be located between the last heater and the boiler. Ordinarily, however, in specifying the energy comsumption of the *turbine alone*, the feedwater is credited with only the heat content of saturated water corresponding to the feedwater temperature leaving the last heater. In a high-pressure installation this heater is generally placed before the most important feed pump, being so located in order to reduce the cost of the heaters even though the volume of water to be pumped is thereby increased.

(e) **Estimates of station heat rates** that may be expected under variable operating and load conditions over long periods of time, say a year, are always necessary when considering a new plant. Since the station heat rate is so dependent upon the performances in the boiler and turbine rooms, either one may be chosen as the starting point in the calculations. Thus one may calculate the generator output expected from burning a unit weight of fuel of known heating value, or he may estimate the amount of heat supplied by the fuel to deliver a unit of energy from the generator. The latter procedure will be the one followed here.

First, the rate of energy consumption E_t of the turbine generator is calculated for the most economical (M.E.) load, as that is the one which the turbine is designed to carry under normal conditions. Then the estimates regarding the performance of the other apparatus in the station enable the station heat rate to be calculated. Thus the fraction E_w of the energy E_t is returned to the feedwater by the feed pumps and the heat absorbed from the fuel is therefore reduced in the proportion $(1 - E_w)/1$. The energy required by the feed pump motors and that needed to operate all other station auxiliaries may be expressed as the fraction E_a of the gross output of the main unit, and this increases the heat rate of the station in the ratio $1/(1 - E_a)$. The efficiency of the boiler unit is estimated for its most economical load, and an *operating factor* is introduced to correct for the losses in both the boiler and turbine rooms due to fluctuations from their respective M.E. loads and for minor miscellaneous losses that will be given later. Therefore the probable rate at which heat must be supplied by the fuel to operate the station, or the *estimated station heat rate*, becomes

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[20] This is discussed on page 264, Part I.

$$\text{Sta. Ht. Rate} = \frac{(\text{Rate of Energy Cons. of Main Unit})(1 - E_w)}{(\text{Boiler-Unit Eff.})(\text{Operating Factor})(1 - E_a)}. \quad (378)$$

This equation will always give the station heat rate in the same units as those used to express the energy consumption of the turbine-generator. Methods of estimating the numerical values of the various factors in this equation will now be considered.

(f) For low pressures the term $(1 - E_w)$ is almost unity and may be taken as such without appreciable error, but for high pressures E_w may be so large that it should be considered. For example, if a turbine is using 10 lb. of steam per kw-hr. with a throttle pressure of 200 lb. per sq. in. abs., and a feedwater temperature of 195 deg. fahr., the use of Eq. (377*b*) gives $E_w = 76(10)^{-6} \times 10 \times 200 \times 0.0166 = 0.0025$, or $\frac{1}{4}$ of 1 per cent, which ordinarily may be disregarded; but if the throttle pressure is 2000 lb., the feed temperature 380 deg. fahr., and the turbine steam rate 7.5 lb. per kw-hr., then $E_w = 76(10)^{-6} \times 7.5 \times 2000 \times 0.0184 = 0.021$ or 2.1 per cent, which is far from negligible. Furthermore, this result is obtained from Eq. (377*b*), which assumes only a 50 per cent recovery ratio, which may be much lower than the actual value.

These numerical values serve to bring out the point that again needs to be emphasized, that the working substance flowing to a turbine is supplied not only with *heat* from the boiler but also with the more expensive form of *energy supplied by the feed pump*, as the latter must be operated by mechanical energy. The rate of *energy consumption* of the *turbine* is therefore a more correct form of expression than "*heat consumption*" or "*turbine heat rate*" and is consequently preferred and generally used in this text. On the other hand, a steam *station consumes heat*, which ordinarily is obtained from the combustion of fuel, and therefore its *heat rate* is a definite and correct form of expressing its performance.

(g) The **operating factor,** as used in Eq. (378), is one that is intended to take care of the station losses due to variable loads on both the boilers and the turbines, the losses due to variation in the temperature of the circulating water, standby losses, and loss of heat from pipe lines. No one can assign a value to this factor for any one station and expect it to hold all of the time, or to apply to any other station; but for a large modern plant these losses may reasonably be assumed to have values somewhat near the following:

	Per Cent
1. Fractional and overload operation of turbine and operation with condenser pressures greater than the normal one (say 1 in. Hg abs.)	3.0
2. Variable load operation of boilers, blow-down, soot blowing, and standby	2.0
3. Heat losses from piping	1.0
4. Leakage, blowing of safety valve, and miscellaneous items.	1.0
Total	7.0

Therefore, for such a station, the operating factor would be 100 − 7, or 93 per cent.

(h) The **energy required by the station auxiliaries,** as represented by E_a, must be known from test, or calculated, before Eq. (378) can be used. The *main auxiliaries* consist of the feed pumps,* condensing-water pumps, draft fans, and the fuel-handling, preparing, and burning equipment; the *miscellaneous auxiliaries* consist of the service-water pumps, bilge pumps, air compressors, cranes, and other small power-consuming apparatus. Some stations have all of their auxiliaries driven by electric motors; others have some or most of their main ones driven by small steam turbines or engines. Although the method of driving the auxiliary affects somewhat its energy consumption, the latter is, however, largely dependent upon the performance of the main unit. For example, if a large turbine is defective and therefore using more steam per kilowatt-hour than it should, the station will have to handle more fuel, the condenser will require more circulating water, and the feed pump will have more work to do. Thus a logical estimate of the main auxiliary energy may be based on the energy consumption of the turbine. The energy for the miscellaneous auxiliaries may also be estimated on the same basis, although these machines are not so dependent thereon as are the main auxiliaries.

In the discussion which follows, the general equations needed will be derived for determining the factors making up E_a, and in connection with them will be given certain numerical values that may be considered illustrative of those possible to obtain in a large station with modern equipment throughout.

First consider the *energy*, E_f, required to operate all of the *equipment*

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* See pages 121–123.

used in any way with the fuel and the products of combustion. This apparatus includes coal conveyors, coal pulverizers, and burners, or stokers, fuel oil pumps and atomizers, also induced- and forced-draft fans. The energy required by this equipment per kilowatt-hour of generator output is, in kw-hr.,

$$E_f = CE_t/[(\text{Boiler Eff.})\ (\text{Operating Factor})\ (10)^6], \quad (379a)$$

in which E_t = energy consumption of turbine-generator at the most economical load, in B.t.u. per kw-hr.

C = energy consumption of all the combustion and fuel equipment, in kw-hr. per million *B.t.u.* liberated in the furnace.

The value of C varies over a considerable range, depending upon the equipment, fuel, and the rate of firing. For gas the value of C is appreciably less than for solid or liquid fuels. For coal or oil a liberal estimate of C is 2, which corresponds to 50 kw-hr. per ton of coal having a heating value of 12,500 B.t.u. per lb.[21] If this estimate be used,

$$E_f = 2(10)^{-6}E_t/[(\text{Boiler Eff.})\ (\text{Operating Factor})]. \quad (379b)$$

Since this result is expressed in kilowatt-hours per kilowatt-hour of generator output, it may also be used as a fraction of the total generator output; and this is also true of Eqs. (380) and (381) which follow.

The *energy consumption* (E_c) of the motors used for the *condensing-water pumps* depends upon the combined pump and motor efficiency (e_p), the total head (z) in feet against which the water is pumped, the quantity of heat (q) in B.t.u. absorbed by each pound of condensing water in passing through the condenser, and Q_c, which represents the amount of heat absorbed by the condensing water, in B.t.u. per kilowatt-hour of generator output. The general expression for this auxiliary energy, in kw-hr. per kilowatt-hour of generator output, becomes

$$E_c = Q_cz/(3413 \times 778 \times e_pq). \quad (380a)$$

For average conditions in central stations fair estimates would be: $e_p = 0.80$, $z = 15$, and $q = 7$, and for such values,

$$E_c = 1.35(10)^{-6}Q_c. \quad (380b)$$

The quantity of heat, Q_c, delivered to the condenser is equal to the energy consumption of the turbine, E_t, less (1) the energy delivered by the generator, (2) the generator losses, (3) the mechanical energy consumed by the turbine oil pumps and main bearings, and (4) heat dissipated from the turbine casing, all per kilowatt-hour of generator output. These losses have previously been considered (in Sect. **274**(e)) sufficient to absorb 6 per cent of the energy delivered to the turbine shaft. Using this amount, then, for each kilowatt-hour delivered by the generator, $Q_c = E_t - 3413/0.94$. Consequently Eq. (380*b*) becomes

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[21] Further consideration of these values will be given later.

$$E_c = 1.35(10)^{-6}[E_t - 3413/0.94]. \quad . \quad . \quad . \quad . \quad (380c)$$

The *miscellaneous auxiliaries* require a small amount of energy E_x, per kilowatt-hour of generator output, that may be estimated over a long period of time as about equal to $\frac{1}{2}$ of 1 per cent. Finally, the part y (=about 5 per cent) of the electrical energy supplied for operating the auxiliaries is lost in transformers and cables in the station.

Thus summing up Eqs. (379), (380) and the other quantities involved, the *total energy required for all of the auxiliaries*, in kw-hr. per kw-hr. of generator output, is:

$$E_a = (E_p + E_f + E_c + E_x) \div (1 - y), \quad . \quad . \quad . \quad (381)$$

in which the computed or estimated values of the various factors involved may now be substituted in order to determine the numerical value of E_a needed in solving Eq. (378).

(i) The **efficiency of the boiler unit,** as used in Eq. (378), means the ratio of the heat absorbed by the working substance from the fuel bed and the products of combustion in any period of time, to the heating value of the fuel fired in the same time. This definition implies that the heating surfaces involved may be distributed in any manner desired between the economizer, air preheater, water-cooled furnace, "boiler surfaces," superheater, and the gas reheater.[22]

In making a comparison of the thermal performances of two different stations, or in comparing the performance of a turbine and an entire station, the boiler-unit efficiencies are of the utmost importance; also, when plant performances are calculated for the purpose of studying the effect of different steam pressures, temperatures, and cycles, the boiler efficiency must be carefully estimated. Some engineers believe that its value should be considered as decreasing somewhat with increased steam pressures, due to the correspondingly higher saturation temperatures of the steam. This view assumes that the gases are always discharged to the stack at a temperature that depends only upon the saturation temperature in the boiler. Other engineers, however, believe that the high-pressure boiler units need not be assumed as having any appreciably lower efficiencies, because the high-pressure installations justify sufficient surface in air preheaters and economizer to keep the stack temperature low enough to obtain high efficiencies. These are questions that are largely decided by the allowable first cost of the boiler-room equipment and will be discussed in later chapters. A steam-generator efficiency of 85 per cent may be assumed as conservative in large modern stations.

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[22] In subsequent chapters the various elements of steam-generating units and their efficiencies will be treated rather fully, but now the only purpose of this definition is to enable a clear understanding of its use in Eq. (378.)

(j) Finally, with the means for estimating all factors in Eq. (378) thus known, it is now possible to apply this equation for determining the probable station heat rate for any proposed plant, more particularly, when there are not available *actual* performance-data from similar plants suitable for this purpose. For example, this equation is particularly useful in investigating the effect of applying higher throttle pressures and temperatures than are now in use. It is also valuable in correcting data from somewhat dissimilar plants, or from similar plants operating under different conditions, so that their performances may be made comparable; and, further, it enables one to determine easily how much a certain percentage change in any one or more factors will affect an actual or estimated heat rate. Thus, if the boiler efficiency should be taken as 80 instead of 85 per cent in the original estimate, then the true station rate would be 85/80, or 106.3, per cent of that based upon the wrong assumption. A similar point is involved with each of the other factors of Eq. (378).

279. Calculated Performances of Stations Using High Pressures and Temperatures.—(a) The method of calculating station performances having been developed in the preceding section, some results of the application of this method will now be given for the different cycles, each over a wide range of pressures and temperatures—some even higher than any that have yet been commercially used. To make the results more significant and useful they are given in the form of curves.

(b) The performances that the authors believe may reasonably be expected from large stations with various steam pressures, temperatures, and cycles are given by the solid curves in Fig. 349, as calculated by De Smaele on the numerical assumptions given in the preceding sections, the more important of which are restated in the figure. The performances of the turbine-generators alone are given by the heavy broken lines in the same charts, the pressure scales of which are logarithmic. The stage efficiency of the turbine is assumed constant when the steam is superheated but decreased in direct proportion to the moisture in the

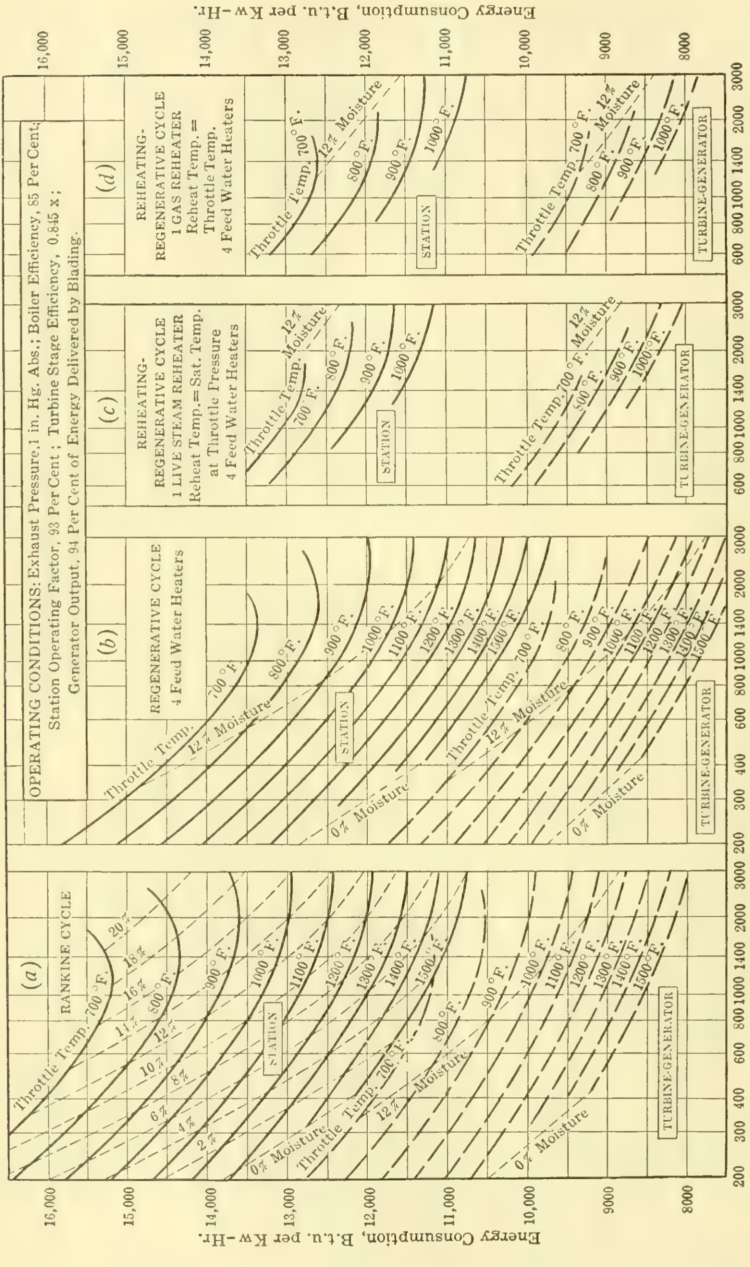

Fig. 349.—Calculated Performance Curves of Large Stations and Turbine-Generators.

lower stages. The assumption of a steam-generator efficiency of 0.85 is probably a conservative one, although that may be considered a debatable point, as mentioned in the preceding section. Should the efficiency be assumed 5 per cent higher, the heat rates would be reduced by an amount ranging from 700 to 1000 B.t.u. per kw-hr. The other assumptions are also quite conservative, hence performances better than those shown by these charts may easily be obtained. The line for 12 per cent exhaust moisture is put on the chart for each cycle as an indication of the upper limit for satisfactory turbine operation. The curves show that if the station operates without any reheating (i.e., on either the Rankine or regenerative cycles), this limit is reached at a throttle pressure of about 350 lb. if the throttle temperature is only 700 deg. fahr., whereas, for a temperature of 900 deg., a throttle pressure of 800 lb. might be used. They also show that with the reheating cycles the pressure limits are much higher, as would be expected. Additional moisture lines are drawn on the chart for the plant operating on the Rankine cycle so as to assist the eye in noting the better exhaust quality that occurs when higher initial temperatures are employed. No allowance has been made in these curves for possible improvement in quality of the steam by separating and extracting the moisture between stages. The moisture curves for the station operating on the regenerative cycle show the same limiting pressures for given temperatures as those for the corresponding Rankine.

For each cycle the turbine-generator curves have been drawn for the best load on the turbine, whereas the curves for the station as a whole include an allowance for the effect of variable load operation, as explained in Sect. **278**(g) in the discussion of the operating factors. The reasons for the differences between the two sets of curves have already been given in connection with Eq. (378).

The curves (*c*) and (*d*), for the *reheating-regenerative plant*, show that with that cycle very high pressures may be used with but a single reheat without approaching the 12 per cent moisture curve, and that excellent thermal performances are obtained without going to temperatures above 900 deg. fahr. The curves (*b*) for the *regenerative plant* with four heaters show what very good thermal economies can be obtained without excessive moisture, even without reheating, if very high pressures and temperatures (above 1000 deg. fahr.) can be used.

Each set of curves in this figure indicates that, if the pressure could be carried sufficiently high, there would be some definite value of it at which the heat rate of the station would become a minimum for each throttle temperature and for each cycle. Thus, for example, with four regenerative feedwater heaters and one live-steam reheater the maximum thermal efficiency of the station would be obtained with a steam pressure of about 2500 lb. if the throttle temperature were 800 deg., and at about 3000 lb., with 900 deg. In this connection special attention is called to the fact that, for any specified temperature and cycle, the steam pressure that gives the best *plant* thermal efficiency is somewhat lower than that which gives the best *turbine* thermal efficiency. For the curves in Fig. 349 this difference is due to the increased feed pump work at high pressures. The difference would be still greater in any plant in which the boiler efficiency becomes lower at the higher pressure.

For a plant operating on the *Rankine* cycle with a pressure of 200 lb. per sq. in. abs. and a throttle temperature of 700 deg. fahr., the curve, if extended beyond the limits of the figure to that pressure, would show a heat rate of 17,200 B.t.u. per kw-hr., and not quite 10 per cent moisture in the last stage of the turbine. Such a station would have nearly 45 per cent greater specific fuel consumption than would one operated on the reheating-regenerative cycle with a steam pressure of 1400 lb. and a temperature of 900 deg. Many other useful comparisons may be obtained from a study of these curves.

(c) The **proper feedwater temperature** to use in order to give the best thermal efficiency of a station operating on the regenerative cycle is always an important matter, but the determination of this temperature is not usually a simple calculation. If the station is one for which the assumptions made in connection with the curves in Fig. 349 are approximately true, then the temperature of the feedwater leaving the last closed heater should be about as shown by the curves in Fig. 350. This set of curves used in connection with Fig. 339 will be of material assistance in analyzing the economic problem involved in selecting the number of heaters and the steam pressure that should be used in a large station in which the operating factor is high and the steam temperature is between 700 and 1000 deg. fahr. When "*open*" or "*direct-contact*" type of heaters are used in place of closed ones, the optimum feedwater temperature becomes somewhat higher than that indicated by the curves in Fig. 350. Some recently built stations are

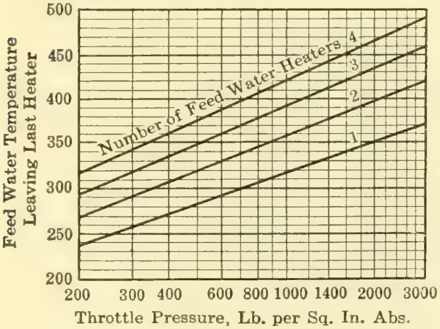

FIG. 350.—Optimum Feedwater Temperature for Highest Thermal Efficiency of Station Using Closed Heaters. (For all steam temperatures from 700 to 1000 deg. fahr.)

using this type of heater, for example,[23] that at South Amboy, N. J., in which 5 open heaters are used and the feedwater temperature leaving the last heater is expected to be 519 deg. fahr. The boiler unit at this plant is designed to furnish steam at a pressure of 1400 lb. and a temperature of 750 deg. fahr.

(**d**) The *calculated performance* of the State Line Station is represented by the diagram[24] in Fig. 351, which shows the results expected when the 208,000-kw. turbine (maximum rating) is carrying its M.E. load of 150,000 kw. The diagram contains many points of interest in addition to those pertaining to the various rates of flow of steam and water. Attention is called to the use of five feedwater heaters and eight condensers for the unit. The steam is reheated once by two live-steam reheaters, arranged in parallel to obtain sufficient capacity. The station heat rate of 13,420 B.t.u. per kw-hr. checks closely the value obtained from the curve in Fig. 349 for corresponding conditions, except that four heaters instead of five were considered in the latter results.

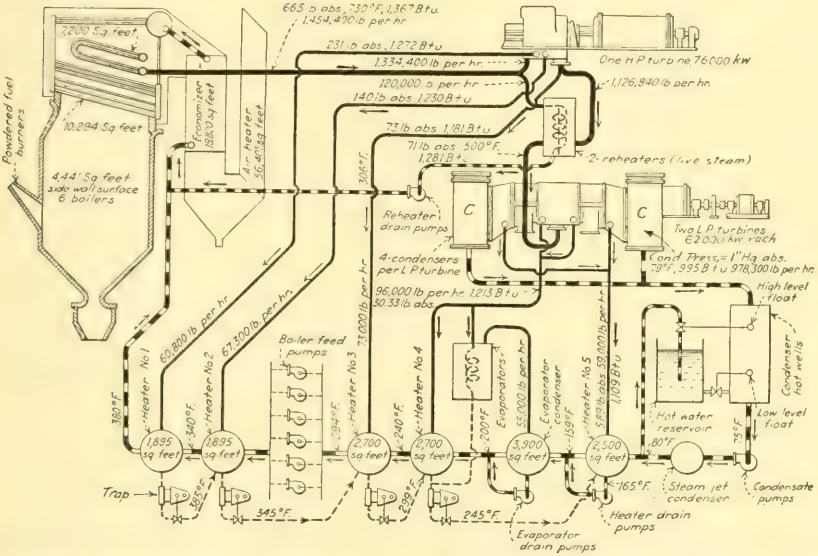

FIG. 351.—Performance Diagram of Triple Turbine Unit Computed for Load of 150,000 Kw.

(Heat rates: Station, 13,420 B.t.u. per kw-hr.; turbine, 9,850 B.t.u. per kw-hr.; auxiliaries 650 B.t.u. per kw-hr. Boiler room efficiency, 85 per cent; operating efficiency rating, 92 per cent.)

(**e**) For stations that use *back pressure turbines* with *high throttle*

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[23] Power Plant Engineering, July 15, 1930, p. 788.

[24] Reproduced from Power, Dec. 3, 1929.

pressures the plant performance may be estimated by a procedure similar to that given in Sect. **278,** but to do this the efficiency of this type of turbine needs to be known for various pressures. The values given by the curves in Fig. 352 by Dr. Melan[25] show what he believes may be expected from turbines having a back pressure of 284 lb. per sq. in. abs. and various throttle pressures up to 2000, the temperature of the steam entering the turbine being sufficient to maintain at the throttle a heat content of 1375 B.t.u. per lb. The latter value means a temperature at the throttle of a little more than 840 deg. at 2000 lb. and about 740 deg. at 600 lb. Two curves are shown in the figure, the lower one being for turbines designed to use half the steam flow of the upper one; the difference between these curves becomes greater as the throttle pressure is increased, due chiefly to the greater percentage of leakage past the blading. As the use of high-pressure steam becomes more and more common in both central stations and industrial plants, the applications of back-pressure turbines will probably become more extensive because of the growing need for exhaust steam for use in industrial processes and for transmission to a considerable distance for district heating.

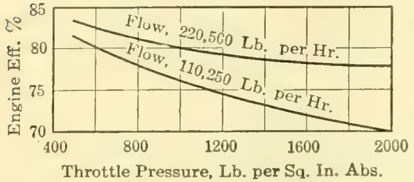

FIG. 352.—Calculated Efficiencies of Back-Pressure Turbines.

Back pressure, 284 lb. per sq. in. abs.; throttle temperature to give h = 1375 B.t.u. per lb.

280. Actual Steam Plant Performances. (a) Before giving actual plant performance figures from tests or yearly operating records, the meaning of the terms *load factor* and *plant factor* needs to be defined. The **load factor** of a station is the ratio of the average power developed by the plant during a certain period of time to the maximum power generated during that period. The maximum power or load is taken for some short time, such as one-half or one hour, and the average power is for some specified period such as a day, month, or year.[26] Thus, for example, if the half-hour maximum is used as a base, the results would be referred to as "half-hour daily," "half-hour monthly," and "half-hour yearly" load factors.

The **capacity plant factor,** also called **plant use factor,** and **capacity factor,** is the ratio of the average load on the plant during some specified time to the rated capacity of the plant. The rated capacity of the plant is measured by the aggregate ratings of the generators installed in the plant.

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[25] Chief Engineer, Siemans-Schuckertwerke, A.G. See Power Plant Engineering, July 15, 1930.

[26] The A.S.M.E. tentative Test Code for complete Steam-Electric Power Plants defines the maximum load as the maximum output generated for one hour.

To show the significance of these two terms suppose a plant contains four 50,000-kw. turbine-generators and that during a certain month of 30 days this plant generated 36,000,000 kw-hr. with a one-hour maximum load of 120,000 kw. Then the hour-monthly load factor of this station would be 41.7 per cent and the corresponding capacity factor would be 25 per cent. Clearly, the higher the yearly load factor on a station the better will be its thermal efficiency, and the higher the capacity factor the smaller will be the fixed charges per unit of energy output.

(**b**) **Actual steam station performance data** as obtained from tests and operating records are not published as frequently as those concerning boilers or turbines alone, but a limited amount of data on station performance of large plants is given in Table XVII. In comparing the performances given in this table special attention should be given to all of the operating conditions and the sizes of the units.

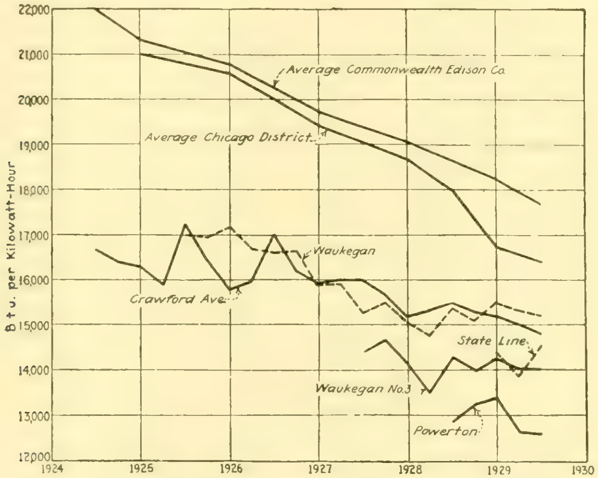

FIG. 353.—Average Heat Rates of Leading Stations in the Chicago District.

(**c**) The **improvement in thermal economy** that has taken place from 1925 to 1930 in large *central station systems* in the Chicago district is clearly shown by the curves in Fig. 353, as presented by Mr. W. S. Monroe, who also shows[27] by Fig. 354 the maximum size of turbine and generator installed in, or planned for, this district from 1901 to 1930.

[27] Power, July 15, 1930.

TABLE XVII

Performances of Steam Stations

Station	Susquehanna (Middletown, Pa.)	Philo (Philo, Ohio)	Columbia (Cincinnati, Ohio)	Richmond (Philadelphia, Pa.)	Trenton Channel (Detroit, Mich.)	Powerton (Pekin, Ill.)	Barton "B" (Barton, England)	Mad River (Springfield, Ohio)
Year of performance	1925	1926	1927	1927	1927	1928	1929	1929
Performance period	72 hr.	1 mo.	1 mo.	1 mo.	1 mo.	4 mo.	1 mo.	1 yr.
Load factor, per cent	....	60.7	60.7	75.6	56.8	59	68.9	42.6
Size of turbine generator, kw.	30,000	40,000	45,000	50,000	50,000	55,000	41,000	20,000
Steam pressure at throttle, lb. per sq. in. abs	333	570	615	406	396	615	375	415
Steam temperature at throttle, deg. Fahr	625	713	725	660	692	725	698	690
Exhaust pressure, in. Hg abs.	2.12	1.37	0.937	1.37	1.15	1.0	0.86	
Number of feedwater heaters.	2	1	3	3	3	4	4	3
Feed water temp., deg. Fahr., leaving last heater	211.5	214	222	308	213	350	304	270
Reheaters	none	1	1	none	none	1	none	none
Efficiency of boiler unit	82.9	84.5	88.1	86.9	87.6	84	80.2	86.3
Station heat rate, B.t.u. per kw-hr	15,900	14,508	12,462	13,700	14,540	12,550	14,545	15,920
Reference	Mech. Eng'g, Apr., 1926	←Engineering, June 6, 1930→				Power, June 10, 1930	Engineering, June 6, 1930	Power, July 15, 1930

The rapid drop in the curve in Fig. 353 representing the average heat rate for the entire district during 1929 and the first part of 1930 should be especially noted. This improved performance is chiefly due to the loads carried by Powerton and State Line.

The information represented by these curves is fairly typical of what might be obtained from many other large steam power systems in this country; but the development in turbines is responsible for only a part of the reduction in station heat rates because the improvements in the other equipment in the station, particularly in the boiler room, and also the better yearly load factors, have also contributed very appreciably to this marked reduction.

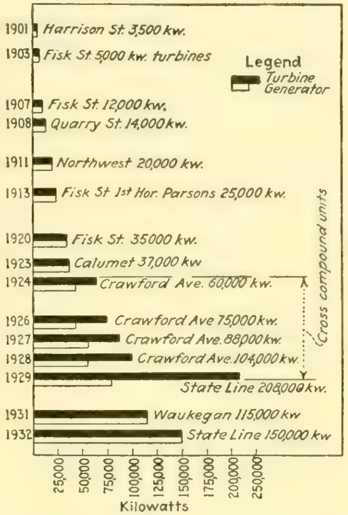

FIG. 354.—Largest Turbines and Generators in the Chicago District.

(**d**) The **mercury-steam**[28] **stations** at Hartford have been in operation a sufficient time now to obtain data from their operating records. The Dutch Point station had its first mercury boiler installed in 1922, and the one in the South Meadow plant was made in 1927. Since the first installation much development work has had to be done on both the boilers and turbines. The performance data of the South Meadow station for a four months' period, in 1930, as presented[29] by Mr. James Orr, are given in Table XVIII. The station heat rate of 10,200 B.t.u. per kw-hr. during this time is an excellent performance; but the fixed charges per kilowatt-hour delivered by such a station are not given. Such information is important, for often the fixed charges are much greater than the fuel costs, especially when there is a low plant use factor.

(**e**) The *complete development* that has taken place in steam central stations since 1900 cannot be easily and fully appreciated by any one, unless he has been directly concerned with it. However, Table XIX, which is abridged from the one given by Mr. F. A. Annett,[30] enables one to see by a glance something of what has been done in this country toward the use of large units, high pressure, and temperature, all of which tend to improve thermal economy. But if steam cannot be gen-

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[28] The mercury-steam cycle was discussed in Sect. **220**, Part I.

[29] Power, July 1, 1930.

[30] Power, June, 1934, p. 344.

TABLE XVIII

PERFORMANCE DATA OF MERCURY-STEAM STATION

Capacity from mercury turbo-generator, kw.	10,000
Steam produced from mercury condensers per hour, lb.	129,000
Average steam temperature leaving superheater, deg. fahr.	735
Average steam pressure in boiler drum, lb. per sq. in. gage	280
Average feedwater temperature to economizer, deg. fahr.	214
Average temperature water to boiler, deg. fahr.	275
Mercury vapor pressure at throttle, lb. per sq. in. gage	70
Mercury vapor temperature at throttle, deg. fahr.	880
Mercury vacuum temperature, deg. fahr.	445
Vacuum in mercury condenser, in. Hg	28.25
Coal burned per hour, lb.	15,800
Heat in coal as fired, B.t.u. per lb.	14,300
Temperature of air at burner box, deg. fahr.	460
Temperature of gas to stack, deg. fahr.	335
CO_2 in flue gases, per cent	14.0
Performance from the start on Feb. 4 to June 4, 1930, inclusive:	
Hours in service	2,032
Coal burned, lb.	26,377,910
Water evaporated, lb.	216,670,000
Output mercury turbine-generator, kw-hr.	16,083,000
Equivalent output from steam, kw-hr.	21,550,060
Total station service, kw-hr.	721,130
Total net output, kw-hr.	36,911,930
Coal rate on total net output, lb. per kw-hr.	0.7146
Station heat rate, B.t.u. per kw-hr.	10,220

erated and used efficiently the station will have poor thermal economy, consequently the efficiency of the boiler unit and the engine efficiency of the turbine are of the utmost importance for any station, regardless of the size of units and the steam pressure and temperature used. The approximate values of the maximum efficiencies attained at various times since 1900 by boiler units and steam turbines are indicated by the curves in Fig. 355. The approximate values of turbine floor space and the velocity of the blade tip in the

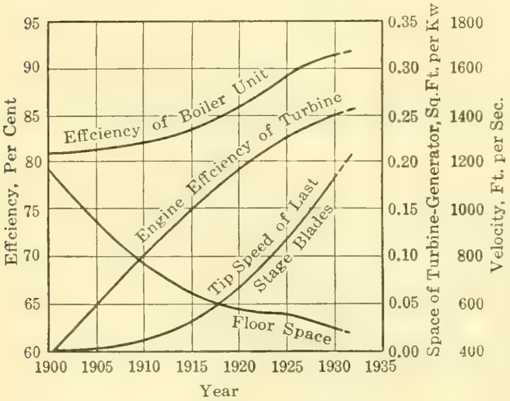


FIG. 355.—Some Developments Since 1900.

last stage of the turbine are also shown in this figure because they have contributed so largely, both in an economic and a technical sense, in the development of the steam-electric central stations from which the people in this country receive about 65 per cent of their electric energy.

281. Selection of Steam Prime Movers.—(a) In some cases the type of prime mover to use for the prevailing conditions is obvious, and in other cases the determination must be based upon a careful analysis. For a large central station which is to deliver electric energy the steam turbine would, of course, be selected. When it is desirable to connect the prime mover directly to a machine which must operate at low speed, and when the use of gearing or other speed-reducing device is objectionable, the reciprocating engine would be chosen. When the amount of power required is small and the thermal economy is at the same time very important, the steam engine may be preferred, especially if the initial pressure of the steam is high. In units of moderate size the thermal performances of engines and turbines may be about equal, and the selection must therefore be based upon considerations other than their fuel consumption.

(b) When the demand is primarily for small amounts of power and there is little or no need for exhaust steam to meet the heating requirements of industrial processes or other applications, the thermal efficiency of the unit becomes an important consideration, especially if fuel is expensive; hence in such case, if the power demand is not large, the most economical type of unit to use may be an efficient steam engine of one of the more complicated types, although its first cost is relatively high. If considerably more steam is needed continuously for heating than is required for generating the power, and if the fuel is very cheap, then an inexpensive arrangement of engine or turbine may be the best form of prime mover to use even though it has relatively low thermal efficiency. If the heating is required only intermittently during the day, or only in certain seasons of the year, some form of engine, or possibly turbine, of intermediate thermal economy, complication, and cost would give the best results.

In such cases the decision as to the best kind of prime mover to use is reached by analyzing each of the possible types as to its determining characteristics (such as its first cost, average maintenance and operating cost, and its general suitability) under the conditions to be met, and then finally selecting the one showing the most favorable results in a comparison.

(c) In general, a unit should be selected to give the greatest return on the cost of operation and the initial investment of the plant as a

TABLE

SOME TYPICAL LARGE CENTRAL STEAM

NAME OF PLANT AND LOCATION	Steam-Generating Units												
Boiler Pressure, Gage, Lb. per Sq. In.	Date of Initial Operation	No. of Units	Max. Capacity, 1000 Lb. per Hr.	Pressure (Gage), Lb. per Sq. In.	Steam Temp. Deg. Fahr.	Boiler Heating Surface, 1000 Sq. Ft.	Water Wall Surface, Sq. Ft.	Furnace Vol., 1000 Cu. Ft.	Max. Liberation, 1000 B.t.u., Hr. Cu. Ft.	Economizer Surf., 1000 Sq. Ft.	Air Preheater Surf., 1000 Sq. Ft.	Type of Firing Equipment	Average Efficiency of Steam-Generating Unit
1200 and Above:													
Edgar............Boston, Mass.	1925	1	143	1200	708	15.6	...	7.6	26.5	11.5	....	....	....
		2	250	1400	750	15.1	1235	10.0	35.4	5.6	33.0	U.S.	84.0
		2	250	1400	750	7.0	1206	10.5	33.7	9.6	29.7	....	....
Deepwater....Penns Grove, N. J.	1930	4	331	1250	725	54.0	2502	19.1	22.6	18.3	46.6	P.C.	88.9
		2	290	....	....	.....	2139	....	....	4.8	52.8	....	....
Deepwater........Houston, Tex.	1924	2	350	1365	850	7.8	1473	13.5	35.0	13.8	31.2	G.	78.5
South Amboy.South Amboy, N. J.	1930	3	280	1400	750	7.2	2036	12.9	27.5	4.4	32.1	P.C.	84.0
Northeast......Kansas City, Mo.	1929	2	230	1400	725	17.0	3050	15.6	21.0	10.9	39.3	P.C.	85.0
Lakeside........Milwaukee, Wis.	1926	4	300	1300	725	23.6	5205	28.6	14.9	....	77.5	P.C.	86.8
Port Washington.Milwaukee, Wis.	.....	1	690	1320	825	44.1	7710	58.0	15.2	....	121.0	P.C.	87.5
Gilbert...........Holland, N. J.	1930	2	250	1400	743	7.9	2194	18.1	20.2	11.9	32.5	P.C.	85.8
Station "A"..San Francisco, Cal.	1931	1	500	1350	750	10.7	1617	15.1	37.9	21.1	46.9	O.&G.	83.6
		2	Reh	eat	...	.....	.....	....	43.3	17.9	51.6	....	....
Station "B"...San Antonio, Tex.	1930	1	215	1518	810	5.0	1011	7.0	40.0	11.9	13.9	G.	75.0
700 to 500:													
Grand Ave.....Kansas City, Mo.	1929	4	230	700	750	15.2	2250	8.6	50.0	9.2	22.1	C.G.	80.0
		1	....	....	....	.....	.....	6.5	....	....	....	....	....
Michigan City.Michigan City, Ind.	1931	3	345	700	775	7.9	2250	14.7	31.8	14.9	42.0	P.C.	85.8
Burlington.....Burlington, N. J.	1932	1	600	730	860	30.3	4816	25.9	30.0	....	41.8	O.	84.4
State Line......Hammond, Ind.	1929	6	450	670	740	10.3	3900	21.0	26.0	19.8	56.4	G.& P.C.	85.0
Crawford Avenue....Chicago, Ill.	1924	12	150	600	750	16.6	.....	6.0	30.0	11.1	10.0	C.G.	78.2
		3	Reh	eat	Boil	ers	.....	....	....	....	....	....	....
		5	200	600	750	17.3	1253 2813	5.8	40.0	7.4	15.9	C.G	78.2
	....	6	200	600	750	17.3	1386	5.8	40.0	7.4	16.0	C.G.	78.2
	....	6	200	600	750	17.3	1786	5.8	40.0	6.9	17.8	C.G.	78.2
Twin Branch....Mishawaka, Ind.	1925	8	150	650	750	94.6	.....	5.1	25.6	8.8	9.5	C.G.	80.0
K. St. Station......Lincoln, Neb.	1930	2	140	600	740	10.5	727	5.8	33.8	3.8	18.1	G.&S.	84.0
Louisiana......Baton Rouge, La.	1930	5	364	640	750	13.8	1843	16.9	31.4	9.8	32.8	G.&O.	....
Philo..................Philo, O.	1924	8	135	650	750	94.6	...	5.4	....	8.8	....	C.G.	81.0
	1929	2	125	650	750	95.6	1597	8.3	50.0	15.8	28.1	C.G.	81.0
	....	4	275	...	....	.....	.....	....	....	....	43.4		
	....	2	350	650	750	43.1	3022	15.2	....	....	58.6	P.C.	81.0
Buzzard Point.Washington, D. C.	1933	2	375	670	835	5.5	3125	12.1	38.5	11.1	47.0	P.C.	86.4
Waukegan........Waukegan, Ill.	1927	4	185	650	725	14.4	.....	....	....	7.1	90.0	C.G.	....
	1930	3	300	650	750	6.2	2900	....	....	17.2	40.1	P.C.	....
	1931	3	500	650	750	10.0	3810	....	....	28.6	64.1	P.C.	....
Stanton............Pittston, Pa.	1927	8	175	675	725	119.7	.....	7.8	43.0	3.0	40.5	C.G.	78.8
Columbia.......Cincinnati, Ohio	1925	8	200	635	730	15.1	425	12.9	21.0	7.2	9.9	P.C.	86.5
Edgewater......Sheboygan, Wis.	1931	2	300	650	825	5.4	2724	10.3	40.0	14.8	40.7	P.C.	86.6
Huntley No. 2.Tonawanda, N. Y.	1930	4	600	500	750	36.8	3301	22.9	34.0	.	90.0	P.C.	86.5
Bremo.........Bremo Bluff Va.,	1931	2	200	525	825	19.8	1838	9.4	25.6	16.0	.....	P.C.	81.0
500 to 400:													
Gorgas Unit 4.......Gorgas, Ala.	1929	2	450	475	725	30.3	3587	24.7	22.0	...	30.6	P.C.	85.0
*Hudson Ave....Brooklyn, N. Y.	1926	4	290	415	700	22.9	1540	...	....	...	43.1	....	....
	1928	8	350	415	720	23.8	1322	....	....	15.0	....	U.S.	85.9
	1932	8	530	415	740	24.5	3846	...	..	22.4	.....	....	....
Ashtabula.........Ashtabula, O.	1930	6	416	400	725	32.6	...	32.6	18.7	23.2	.....	P.C.	...
Avon.................Avon, O.	1926	8	340	400	725	30.6		30.6	15.5	21.1	...	P.C.	...
Gould St.........Baltimore, Md.	1927	2	500	440	735	26.4	3737	21.0	31.0	...	26.8	P.C.	84.0
Delray No. 3......Detroit, Mich.	1929	5	340	400	725	23.9	2044	13.8	35.0	6.0	29.9	U.S.	84.4
Trenton Channel..Detroit, Mich.	1924	13	360	400	725	29.1	1220	25.0	18.0	19.0	.....	P.C.	87.0
East River..New York City, N. Y.	1926	6	250	400	725	14.8	4134	....	...	...	28.9	P.C.	85.9
		3	1000	....		60.7	7345	...			82.7	....	....
Binghamton..Johnson City, N. Y.	1927	3	150	448	696	10.6	1250	8.4	25.0	9.2	12.0	P.C.	77.5
Richmond......Philadelphia, Pa.	1925	12	200	400	725	15.7	597	7.8	33.0	7.5	22.1	U.S.	86.2
	1935	2	600	425	850	36.2	4648	26.9	30.0	...	48.0	P.C.	86.2
Long Beach....Long Beach, Cal.	1928	6	425	455	775	34.2	3240	19.8	27.0	....	51.2	O.&G.	84.0

U.S., underfeed stoker; P.C., pulverized coal; O.&G., oil and gas; C.G., chain-grate stoker; G.&S., gas and stoker; S., stoker; P.P., pulverized petroleum coke; P.L., pulverized lignite; C.C., cross-compound turbine; S.C., single-cylinder turbine; T.C., tandem-compound turbine; V.C., vertical-compound turbine; B.P., back-pressure turbine; B.T., bleeder turbine; D.E., double-extraction non-condensing turbine; A., high-pressure plant

XIX

PLANTS IN THE UNITED STATES

Abridged from
Power, June, 1934

Fans				Final Feedwater Temperature, Deg. Fahr.	Air Temperature Leaving Preheater, Deg. Fahr.	Turbines							Condenser Tube Area, 1000 Sq. Ft.	Auxiliary Power, Per Cent of Total Gen. Capacity	Yearly Load Factor, Per Cent	Yearly Plant-Capacity Factor, Per Cent
Forced Draft		Induced Draft														
No. per Unit	Capacity Each, 1000 C.F.M.	No. per Unit	Capacity Each, 1000 C.F.M.			Number	Arrangement	Capacity Each, 1000 Kw.	Capacity High Press. Unit, 1000 Kw.	Throttle Press. (Gage), Lb. per Sq. In.	Throttle Temperature, Deg. Fahr.	Regenerative Stages				
1	80.0	1	150.0	.....	.....	....	.....	38	3.4	.....	.....	3	45.0	.....	.....	.....
2	47.5	2	66.0	308	410	3	C.C.	38	11.8	1250	712R	3	45.0	6.25	50.4	.....
2	47.5	2	70.0	.....	454	....	.....	65	13.8	1250	.....	3	50.0	.....	.....	.....
1	75.0	1	150.0	415	470	2	C.C.	53	11.0	1240	715R	4	42.5	6.4	73.0	82.0
....	......	....	......	.....	.....	1	S.C.	.....	12.5	1240	715	Exh	aust pr	essure	400 lb.	max.
2	55.0	2	105.0	330	320	1	A.	.....	12.0	1282	710R	....	......	6.8	62.0	60.0
2	40.0	2	62.0	520	477	2	V.C.	25	5.3	1250	750R	5	20.5	7.0	70.3	35.2
1	90.0	1	153.0	325	350	1	A.	.....	10.0	1300	725R	....	......	......	70.7	.....
2	47.5	2	92.0	300	635	4	A.	.....	7.7	1230	750	....	......	6.5	61.2	61.2
2	112.5	2	185.0	398	592	1	T.C.	80	......	1230	825R	5	52.5	......	......	.....
1	76.5	1	153.0	303	405	1	C.C.	55	11.8	1248	735R	3	31.6	7.5	63.0	53.0
1	135.0	1	240.0	425	472	2	V.C.	50	12.5	1230	750R	4	34.0	16.8	17.5	14.4
1	165.0	1	190.0	.....	474	....	.....	.....	......	......	.....	....	.....	.....	.....	.....
1	70.0	1	110.0	371	342	1	A.	20	8.0	1250	810R	3	22.5	17.0	42.2	26.3
1	83.5	1	156.0	300	340	2	B.P.	10	......	600	740	1	......	......	13.0	....
							S.C.	35	......	600	740	2	45 0	8.0	55.0	....
1	96.0	1	178.0	300	430	1	S.C.	68	......	675	750	4	50.0	6.3	43.1	45.8
1	168.0	1	335.0	340	460	1	A.	.....	18.0	650	846	....	......	......	60.0	60.0
2	60.0	2	110.0	355	400	1	C.C.	208	76.0	660	730R	5	176.0	5.2	60.0	60.0
2	40.0	1	115.0	300	230	1	C.C.	51	16.0	550	750R	4	56.0			
....	......	....	......	.....	.....	1	C.C.	62	17.0	550	750R	4	75.0			
2	45.0	1	125.0	300	230	1	C.C.	52	30.0	550	750R	4	65.0	4.7	64.0	45.0
....	......	....	......	.....	.....	1	C.C.	77	28.0	550	750R	4	70.5			
1	90.0	1	125.0	300	230	1	C.C.	91	33.0	550	750R	4	70.5			
1	90.0	1	125.0	300	230	1	C.C.	104	45.0	550	750R	4	90.0			
1	70.6	1	130.0	200	270	2	S.C.	40	......	600	710R	2	25.0	5.5	......	.....
1	56.0	1	80.0	212	350	2	B.T.	.....	6.0	550	725	3	7.4	11.8	50.5	61.8
1	109.0	2	84.0	350	439	3	B.P.	15	......	600	735	Exh	aust pr	essure	140 lb.	gage
1	70.6	1	130.0	200	.....	2	S.C.	40	......	580	710R	2	45.0			
2	50.0	1	175.0	290	415	1	C.C.	165	53.0	580	710R	4	110.0	5.7	60.0	60.0
					415											
2	60.0	1	210.0	290	460	....	......	.....	......	......	......	....	......			
2	50.0	2	84.0	364	425	1	S.C.	35	......	......	......	3	30.0	4.8	......	.....
2	32.9	1	112.8	265	315	1	S.C.	50	......	625	725	3	57.0	5.8	62.3	16.8
1	76.5	1	145.5	365	380	1	T.C.	65	......	625	750R	4	63.0	5.8	70.6	44.7
2	68.6	1	243.3	325	490	1	T.C.	115	......	625	750R	4	80.0	5.8	58.2	48.2
2	33.0	2	56.0	210	480	2	S.C.	50	......	645	725R	2	33.0	4.5	88.0	87.0
1	85.0	1	100.0	233	278	2	C.C.	63	16.0	600	720R	3	48.0	6.5	56.0	52.0
2	43.5	1	150.0	305	347	1	S.C.	30	......	650	825	4	24.0	6.3	35.8	50.3
2	95.0	2	105.0	356	342	2	S.C.	80	......	425	750	4	45.0	6.7	25.2	27.6
Diphenyl oxide used in air preheaters				380	....	2	T.C.	15	......	450	825	4	13.0	5.0	36.0	28.0
1	166.0	1	358.0	385	.....	1	S.C.	60	......	425	725	4	70.0	5.0	S.	S.
2	60.0	2	90.0	.....	275	1	C.C.	80	......	410	690	4	80.0			
1	120.0	2	105.0	235	.....	2	C.C.	110	......	410	706	2	85.0	4.0	49.1	33.5
1	200.0	2	155.0	.....	.....	2	T.C.	160	......	410	730	2	101.0			
....	......	1	327.0	330	.....	3	S.C.	50	......	375	700	2	57.0	4.3	......	.....
....	......	1	195.0	327	.....	4	S.C.	35	......	375	700	1	42.0	4.4	......	.....
2	75.0	2	135.0	365	450	2	S.C.	36	......	410	725	4	30.0	5.8	58.4	49.1
2	65.0	1	223.0	300	325	3	S.C.	50	......	375	725	4	60.0	4.6	48.8	45.8
2	13.0	1	193.0	225	.....	6	S.C.	50	......	375	725	3	51.8	4.6	45.4	29.1
1	60.0	1	150.0	300	520	2	S.C.	60	......	385	735	4	47.5	7.5	46.2	40.6
2	125.0	2	260.0	.....	520	1	T.C.	160	......	......	......	4	90.0	......	......	.....
1	28.0	2	54.0	214	344	1	S.C.	30	......	385	687	3	22.0	6.2	65.0	38.4
1	60.0	2	45.0	290	344	2	T.C.	60	......	375	675	3	70.0	4.7	61.1	61.1
2	102.5	2	190.0	325	507	1	T.C.	165	......	375	825	4	13.0	3.0	50.0	50.0
2	53.0	1	180.0	387	400	2	T.C.	100	......	410	735	4	80.0	3.0	54.3	18.9

superimposed on medium-pressure plant; R., steam reheated before going to low-pressure turbine; S., standby service.

* In addition to units shown there are 12 boilers and three 50,000-kw. turbines operating at 275 lb. and 600 deg. F. steam, making the station capacity 770,000 kw., the largest in the world.

whole, considering such factors as were given in Sect. **269.** The choice depends largely on the operating conditions, which include such factors as the steam pressure and temperature, the back pressure (as fixed by the supply of condensing water available or by the pressure needed for heating purposes), the cost of fuel, the cycle through which the working substance is carried, and the use factor. The diversities encountered in the different plant designs and local conditions cause wide variations in the factors involved.

When there are varying loads to be met, the *size* and, therefore, the number, of the units to be used in the plant should be determined with respect to the base load and peak loads, and no unit should be so large that its failure at the time of maximum demand will throw on the other units a burden beyond their emergency capacities.

In many plants the annual *cost of fuel* is less than the yearly *fixed charges* on the investment, i.e., the amount set aside annually for the retirement of capital (amortization) or eventual replacement of equipment, and the amount paid annually for taxes and insurance and maintenance, the sum of these being a certain percentage of the initial cost of the plant. As *small plants* are inherently more costly per unit of capacity than are large ones, equipment of the less complicated and less expensive kind is usually selected for them even at a sacrifice of thermal efficiency. Hence small plants generally do not show as good performances as those given in the preceding sections for large central stations.

(d) The foregoing discussion shows that the proper selection of kind and size of unit under different conditions involves many factors. The previous treatment of steam turbines and their performances may well be supplemented by additional reading, such as the paper[31] presented by Professor A. G. Christie at the 1930 World Power Conference.

Additional treatment of the suitability of different types of the reciprocating steam engine for various purposes will be given in the next chapter.

[31] Economic Considerations in the Application of Modern Steam Turbines to Power Generation, in Mech. Eng., July (p. 673) and Aug. (p. 757), 1930.

CHAPTER XXIII

DETERMINING FEATURES OF PRIME MOVERS

282. (a) Some types of prime movers are so superior to all others in certain fields of work that they are used almost exclusively in them, but in other applications nearly all types may be found and considerable study may be required to decide which one is most suitable in any given case. Thus, for example, the steam turbine is supreme in large central stations, the reciprocating steam engine in large locomotives, and the internal combustion engine in automotive transportation. On the other hand, all of these types of heat engines are extensively used in marine work, and also in various kinds of small stationary plants. This chapter will give a brief discussion of the main features that determine the type of prime mover best suited for different applications.

(b) The various *types of steam turbines* used in stationary plants have been considered rather fully in Chapter XXI, so there is need here for only a short additional discussion regarding their features that fix the type and construction best suited to other work. On the other hand, the treatment of the reciprocating steam engine has thus far not included a number of features that have an important bearing on its suitability for different fields of work and therefore a considerable portion of this chapter will be dealing with them.

283. Determining Features of Steam Engines.—(a) Some of the mechanical features that should be incorporated in engines are dependent largely on the conditions under which the machines are to operate and on the results that must be produced. In some cases the engine must be low in initial **cost,** thus of simple construction, and be one that can be operated by low-salaried, inexperienced attendants; consequently it must be as nearly " fool proof " as possible and its thermal efficiency becomes a secondary consideration. Such engines will have simple parts that are made of the cheaper materials, using the less costly manufacturing processes and the minimum of labor. Under other circumstances low first cost and lack of complication may be of less importance than good thermal results, hence the engine may be constructed of parts of greater complexity and be made of better materials and with finer workmanship. High-speed, short-stroke engines are smaller and less expensive than low-speed, long-stroke ones; also they are usually of simpler construction and require smaller foundations and less space. Further, if they are "direct-connected" to electric generators or other machines, the apparatus driven is also generally smaller and cheaper because operated at higher speeds. Simple engines are of course less costly than multi-cylinder ones; and the tandem arrangement is less expensive than the cross-compound. When the load varies widely and suddenly, or when the conditions are otherwise severe, the engine must be of massive construction, and is then of the *heavy duty type*; otherwise a lighter and less costly construction may be used.

(**b**) The **rotative speed** of an engine has an important influence on the size and cost of equipment for a given output, as well as on the stresses and vibrations that may occur; this speed also often places a limitation on the kinds of *valves* and *valve gears* that may be used. Certain mechanisms, such as trip cut-off valve gears, will not function with reliability at high speeds of operation, whereas they may be very desirable for low-speed engines. The term "*speed*" will hereafter be understood to mean rotative speed unless otherwise qualified. This speed, N, in r.p.m. and the length of stroke, L (twice the crank radius in feet), determine the **average piston speed,** S, in feet per minute, since $S = 2LN$. With the same speed and weights involved, the longer the stroke of the piston the greater will be the inertia forces of the reciprocating parts; and with a given weight and stroke, increasing the rotative speed will cause greater inertia and centrifugal forces. These forces, which vary as N^2, not only induce stresses in the parts affected, but may cause undesirable vibrations; hence there are limiting combinations of L and N which should not be exceeded. Usually the smaller, short-stroke, high-speed engines (say 4 in. × 5 in.)[1] have a piston speed of about 500 ft. per min., and with large, long-stroke, slow-speed engines (say for one 18 in. × 48 in.), S is about 700 ft. per min. The speeds mentioned are not limitive values, however, for even double these speeds are sometimes used in special cases. Having fixed on suitable values of any two of the factors S, L, and N, the other of course follows.

(**c**) When **high steam pressures** are used, not only must the parts have proper strength, but on large engines it may be necessary to make allowance for the increase in the diameter and length of the cylinders, the stretching or compression of rods and frames, and the distortion of other parts caused by the pressure and temperature. Valve gear of the types which require very accurate adjustment, and the clearance between the piston and cylinder head, may be seriously disturbed by these distortions if proper provision is not made for them. Long engines may vary in length a quarter inch or more during each cycle. With high pressures, elements of inherent structural strength and rigidity should be adopted when otherwise feasible; and the valves should be of types which are nearly frictionless and without wear, and not subject to distortions which would permit leakage or prevent proper functioning.

---

[1] It is always customary to give the piston diameter first and its stroke second.

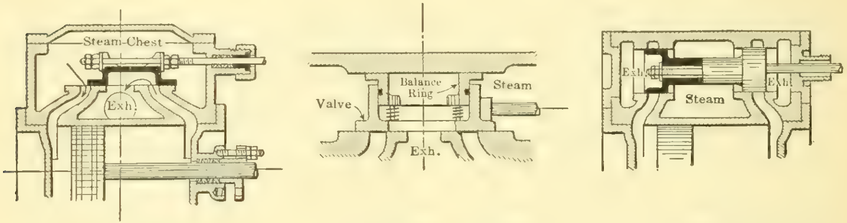
(**d**) If steam of **high temperature** is adopted, the effectiveness of the cylinder lubricant may be decreased thereby; this may, in turn, interfere with the proper functioning of the piston, piston rod, and the valve, and may result in undue wear of these parts, especially of the latter. If such a temperature is used, it may be preferable to select for these elements those types that require little or no lubrication. To prevent the piston from being in contact with the cylinder walls, and therefore requiring lubrication, the piston rod can be extended through a stuffing box in the outer head of the cylinder and have its end carried by a special cross-head which slides on a suitable guide. When such a "tail rod" and "tail rod guide" are used, only the piston rings are in contact with the cylinder surface, and they exert but little pressure. The valve can be of the type that lifts off of its seat without rubbing.

When high temperature steam is used, provision should also be made for the inequalities of expansion of different parts of the engine, and of the various elements in the same part, so as to avoid serious stresses and distortions as well as faulty operation resulting therefrom. This applies particularly to the valves and to their seats and guiding surfaces, and is especially important with large engines.

(**e**) Regardless of the conditions under which an engine operates, the main parts, with the exception of the cylinder, are of much the same general design, since the modifications imposed by special conditions are relatively slight, although often important. But the *kind of valve* used may of necessity have to be selected with reference to the special demands on the engine; and this, in turn, may have a marked influence on the design of the cylinder and certain other parts, as well as on the operation of the engine. Hence the valves will be given further consideration.

284. Valves.—(**a**) Not only must the valves of an engine be designed with respect to such factors as pressure, temperature, friction, and lubrication, but, in addition, the type used may have an important influence on the arrangement of the cylinder of the engine and on the kind of governor adopted. Furthermore, the kind of valve employed may seriously affect the thermal efficiency of the engine, since some valves necessitate the use of long ports which result in large volumes and extensive surfaces in the clearance space and also may cause the entering steam to flow over surfaces that were cooled during the exhaust period, whereas other kinds of valves may minimize these detrimental features. The influence of some of the more common types of valves on these factors will next be considered.

(**b**) The **simple D-valve,** shown at (*a*) in Fig. 356, has the full, steam-chest pressure on its back, and exhaust pressure over most of its under side. The resulting unbalanced pressure is therefore very great and calls for rigid construction. It also causes considerable friction to occur between the valve and its seat. This friction not only wastes power and causes wear, but may put a serious drag on the governor if the valve operation is controlled by such a device. By making the valve short, its weight, friction, and wear can be greatly reduced; and the two latter can be decreased further by using a balance ring, such as that shown at (*b*), or some similar device, to prevent the steam from acting on the greater portion of the back of the valve. The D-valve involves the use of long ports and large clearances in the cylinder, especially if the valve is short; it is relatively heavy, particularly so if provided with certain types of balancing devices. With such valves the steam enters the cylinder by the same passage through which the cooler exhaust steam of the previous cycle was discharged, hence there results a wasteful interchange of heat between the vapor and the walls; furthermore, the exhaust chamber jackets a portion of the exterior of the cylinder from which the outflowing steam abstracts heat. The D-valve is a very simple valve, but obviously is suitable to use only when economy of performance is a secondary consideration, and when the prevailing pressures and temperatures of steam are relatively low, e.g., below 150 lb. per sq. in. and 400 deg. fahr., respectively. Its arrangement is such, however, that should the cylinder happen to contain water in an amount more than sufficient to fill the clearance space, the valve will lift at the end of compression, thus affording a relief without which some part of



(*a*) Simple D-Valve. (*b*) Balanced D-Valve. (*c*) Piston Valve.

FIG. 356.

the engine would probably break, as water is substantially incompressible at common cylinder pressures.

(c) The **piston valve** shown at (*c*), which may be considered to be a D-valve in cylindrical form, is balanced as regards steam pressures, and hence is nearly frictionless and requires but little lubrication. It can be of light weight and may be made long so that the cylinder passages may be rather short and direct. It is symmetrical and hence distorts uniformly with temperature changes. It has been used with temperatures as high as 600 deg. fahr. Further, a valve of this type may be easily arranged as an " internal valve," as shown in Fig. 356(*c*), i.e., to take steam from its inner edges and exhaust at its ends, in which case the jacketing of the cylinder by exhaust steam may be largely or wholly avoided. But the ordinary piston valve will not give relief if there is an excessive amount of water in the cylinder; hence other means must be provided for this. *Cylinder relief valves,* which may be somewhat similar to boiler safety valves, are often attached to the clearance spaces for this purpose.

(d) Rectangular balanced **valves of the Sweet type,** such as that shown in Fig. 357, are, in effect, rectangular piston valves that work between a "pressure plate" or "balance plate" and the valve seat. Such valves are commonly made multi-ported, the one shown in the figure having two separate passages through which the steam may flow into the cylinder ports. This arrangement permits quicker opening and closing of ports and shorter valve-travel with less throttling than a single-ported valve. The balance plate and valve may be arranged to lift when there is water in the cylinder; but, under high temperatures, they may warp and cause trouble. The valve may be of light weight and offer little resistance, due to friction and inertia.

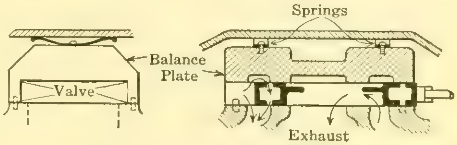


Fig. 357.—Sweet Type of Valve.

(e) Valves of the foregoing types control all four of the valve events, i.e., admission, cut-off, release and compression; hence an engine would have but one of these valves per cylinder, and consequently, such a prime mover is termed a *single-valve engine.* These valves can be operated satisfactorily with rapid motion and hence may be used on *high-speed, short-stroke engines.*

(f) Sometimes a valve is used to control only one, two, or three events, in which case two or more valves are used on a cylinder, the engine then being of the **multi-valve type,** such as will now be considered. The valves may be either of the slide, oscillating (Corliss), or poppet types.[2] In some cases one slide valve may control admission, release and compression (the " fixed " events, which preferably should not vary) and another valve on its back may control the cut-off. The second valve is called a **riding cut-off valve.** Formerly such arrangements of valves were used extensively on medium-speed engines and on steam cylinders that drive pumps and compressors, but now their application is rather limited. Other arrangements of multi-valve engines will be discussed in the paragraph immediately following. Since engines using multi-valves are more complicated than those having single valves, the former are ordinarily more expensive than the latter.

(g) The most widely used of the multi-valve arrangements is that found on the Corliss type of engine, which has four oscillating valves to a cylinder. Each of these **Corliss valves** controls but one event and is located in a separate transverse bore, as in Fig. 358. The valves have cylindrical surfaces and are oscillated through only a small angle. As they are not symmetrical and require lubrication, a steam temperature of 500 deg. fahr. is considered as about the upper limit to use with them. With such valves the steam and exhaust passages in the cylinder are very short and direct; hence the clearance volumes and surfaces are relatively small, the entering steam does not come in contact with port surfaces over which the exhaust steam has passed, and the exhaust steam may be kept from jacketing the cylinder. Thus, the thermal efficiencies secured with these valves are better than those obtained with any of the other valves so far discussed. The inlet valve can be arranged to lift to give water relief. In the older form of Corliss engine, which has had very extensive application, a **trip cut-off** gear

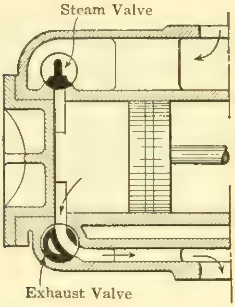


FIG. 358.—Corliss Valve Arrangement.

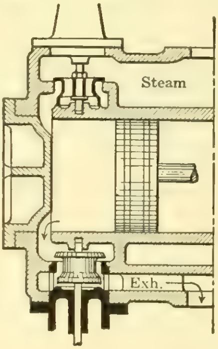


FIG. 359.—Poppet Valve Arrangement.

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[2] The rotating type is seldom used.

is used, the disengagement of the admission valve being accomplished by a simple mechanism (to be described later) which offers little resistance to motion, and hence can be controlled by a simple "flyball governor" which has but little strength. As the tripping mechanism may not function properly under rapid motion, these engines are of necessity of the low-speed type, and hence are made with long strokes. They are usually referred to as *low-speed*, or *trip cut-off, Corliss engines.*

(h) In a more recent application, the Corliss valves are controlled positively at all times, there being no tripping of the admission valve, and the speeds can therefore be the same as those of the high-speed, single-valve engines. This arrangement of engine is known as the *high-speed Corliss,* and it usually has a relatively short stroke. These engines give about as good thermal performance as is obtained with the trip cut-off type.

(i) The **poppet valve** is another type of valve that may be used on multi-valve engines. The poppet valve used on steam engines is commonly of the **double-beat type,** as shown in Fig. 359, which is made hollow and closes against two seats. When this valve lifts from its seats, there are two separate annular passages through which steam flows into the cylinder port, instead of only one such passage as in the single-beat valve. The advantages of the double-beat valve over the single-beat are the very much smaller force required to lift it, since the differential steam pressure acts on a much smaller area on it when closed, and also the smaller throttling that occurs when the valve is opened, due to the larger cross-sectional area of flow for a given lift. The valve is of light weight, easy to lift, has no rubbing surfaces and therefore requires no lubrication, and being symmetrical, changes shape uniformly under high pressures and temperatures. It is suitable for use with steam having fairly high pressure and superheat (say to a temperature of 600 deg. fahr.), but must be designed, constructed, and operated with great care. These valves can be so arranged with respect to the cylinder as to necessitate only the minimum of clearance space and surface, and with them the exhaust steam can be kept from jacketing the cylinder. With such an arrangement the thermal results are therefore somewhat better than with engines using valves of the other types.

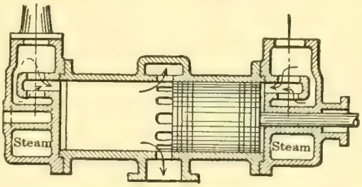

FIG. 360.—Elements of a Uniflow Engine.

(j) The **exhaust valve in uniflow engines** is the *piston itself,* which functions with respect to the exhaust ports located at the middle of the cylinder. The steam or inlet valves, controlling admission and cut-off, are located at the cylinder ends and may be of any of the types considered, although the poppet and piston types are the ones commonly used. It is apparent from Fig. 360 that head-end release, for example, will occur when the left end of the piston uncovers the left edge of the central exhaust port, the piston moving to the right. Since head-end compression occurs when the piston again reaches this position in moving toward the left, compression will begin very early in the return stroke. The operations at the other end of the cylinder are, of course, similar. The principal advantage gained by this early compression is the high terminal temperature produced, which aids in preventing heat transfer and condensation during admission of the live steam. On the other hand, this early compression, combined with the early cut-off, results in low mean effective pressures, and this requires using a piston and cylinder of relatively large diameter for a given power capacity. Also, with a high back pressure, this early compression will produce terminal pressures which exceed the admission pressure unless sufficiently large clearance volumes or auxiliary exhaust valves are provided.

(k) On locomotives and marine engines, as with stationary engines, it is possible to apply any of the valves mentioned, but usually the service conditions require the use of the simplest forms, the piston valve being the one most commonly adopted.

(l) The valves of an engine are driven by mechanisms called **valve gears,** of which there are numerous arrangements, each being best suited for certain purposes. Engines which operate only in one direction have simpler forms of gears than those which must run in either direction. For the latter, some type of *reversing valve gear* must be used. Valve gears will be discussed in Chapter XXV.

285. Types and Applications of Engines.—(a) A **throttling engine** is one in which the pressure of the steam entering the cylinder is regulated by a *throttling governor* in such manner that the power output at all times will meet the variable demand and yet the desired rotative speed will be maintained. Such an engine, the simple elements of which are shown in Fig. 361, always has a fixed cut-off which occurs late in the stroke, to provide for overloading, and the steam is usually considerably throttled when the load is normal; consequently, this type of prime mover is inherently uneconomical thermodynamically, especially when not heavily loaded. However, it is commonly used when low first cost, simplicity, and convenience are more important than good thermal efficiency. It usually has a D-valve, perhaps balanced, or a piston valve, and when cheapness is important, it may have an open type of frame and other parts that are of simple design. This engine is commonly operated with steam having low pressure, usually considerably less than 150 lb. per sq. in., and with no superheat. It is used for driving fans, stokers, and other power house auxiliaries, for operating machinery in industrial plants and for many other purposes, especially when the exhaust steam can be utilized for heating feedwater, for warming buildings, or for furnishing heat for manufacturing processes. When the only steam available has high pressure and high superheat it can be rendered suitable for these engines by passing it through *pressure-reducing valves* and *de-superheaters*. There are many manufacturers who make both horizontal and vertical engines of this type.

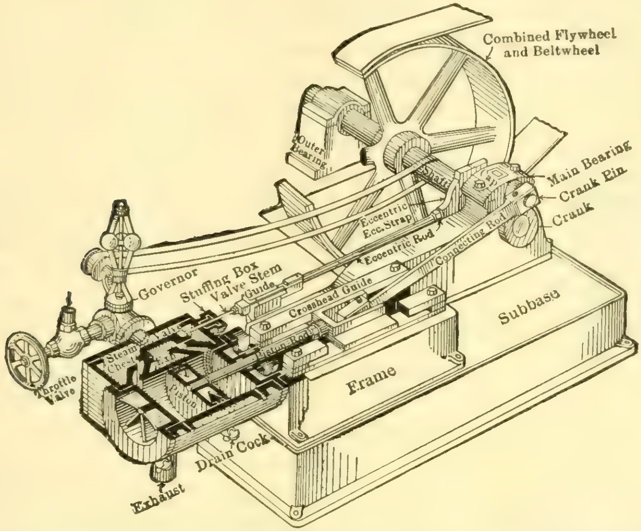

Fig. 361.—Elements of a Simple Throttling Engine with Belt Drive.

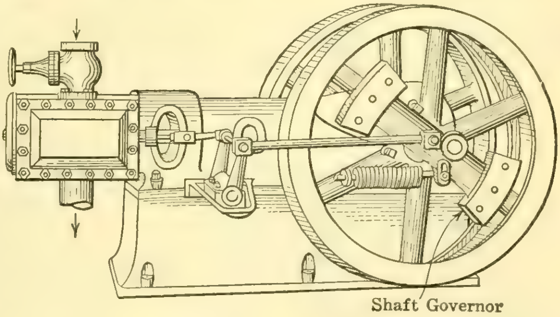

Fig. 362.—High-Speed, Single-Valve, Center-Crank Engine with Shaft Governor.

(b) **High-speed engines,** Figs. 362 and 363, which use but one balanced, single- or multi-ported D-valve, a Sweet valve, or a piston valve, driven by a powerful automatic *shaft governor*, have been built in large numbers for many years in both the horizontal and vertical arrangements. Having lengths of stroke that are relatively short, often not more than equal to the diameter of the cylinder, they are comparatively small and compact and hence usually cost less than the other commercial types of reciprocating steam engines, with the exception of the throttling engine. They are regulated as to power output by changing (automatically) the point in the stroke at which cut-off occurs; hence, especially at light loads, have better thermal efficiencies than does the throttling type of engine previously considered. They have been used very extensively for driving direct-connected electric generators and other high-speed machinery. These engines are often called "*high-speed, single-valve engines*" (H.S.S.V. engines) or more fully, "automatic, high-speed, short-stroke, single-valve engines." Usually they are better built than the throttling engines, have frames that enclose the crank, connecting rod, and crosshead, thus protecting these parts from dust and also preventing the throwing of oil, and have automatic or semi-automatic oiling systems. The commercial sizes of these engines usually range from a few horsepower to 500 hp., the speeds being from 500 r.p.m. for the smallest to 150 r.p.m. for large ones. They find greatest application in the smaller sizes.

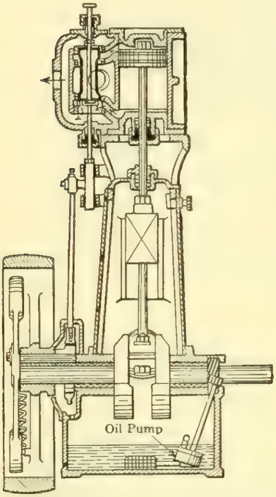

Fig. 363.—A Small Vertical H.S.S.V. Engine.

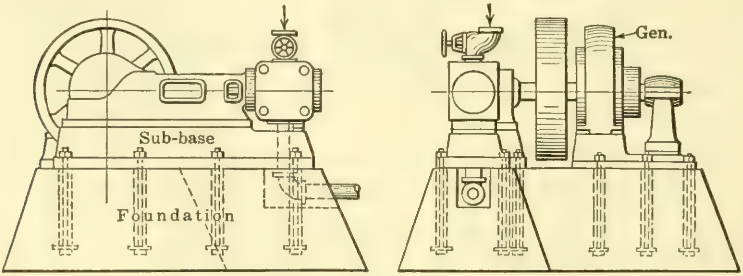

Fig. 364.—Outline of H. S. Corliss Engine, Direct-Connected to an Electric Generator.

(c) The **high-speed Corliss engine,** Fig. 364, differs structurally from the H.S.S.V. machine mainly as to its valves and valve gear. Its greater complication of mechanism may add slightly to its cost, but under suitable conditions this increase in initial expense may be offset by the better operating results. The performances of these machines compare favorably with those of the trip cut-off Corliss engines, but, being operated at higher speeds, they may be expected to wear somewhat more rapidly. The commercial sizes commonly vary from 50 to 800 hp. or more.

(d) **Medium-speed engines** generally have strokes from one and one-half to twice the diameter of cylinder bore in the smaller sizes, and because of this greater length they are operated at rotative speeds that are somewhat less than is used with short-stroke engines. Generally, these engines have been built only in sizes that are somewhat larger than those of the H.S.S.V. type, although the two kinds overlap considerably, and they ordinarily have two or more positively-driven slide valves, commonly of the riding cut-off type, with single- or multi-ported valves. Formerly they were used extensively in power plants, but now the demand for them has greatly decreased, largely because turbines and internal combustion engines have often proved to be more suitable in the fields in which they were once applicable.

(e) Of the **low-speed type** of engine, the most representative is the **trip-gear Corliss** shown in Fig. 365. Because its rotative speed is inherently limited to 100–150 r.p.m., or often less, by the type of valve gear used, the strokes are made relatively long, the ratio of stroke to bore being usually from 2 to $3\frac{1}{2}$ or more. Long-stroke engines of this kind have given very satisfactory service for many years. They are somewhat more costly than the other types so far considered, but are more economical in the use of steam than engines having slide valves. The commercial sizes ordinarily range from 50 hp. to 1500 hp. or larger. Large engines, in general, must necessarily operate at relatively low rotative speeds, otherwise the piston velocities would become excessive; and, although the stroke may be long, the ratio of its length to the diameter of piston may be small. The classification of an engine as "low-speed" is dependent on the limitations placed by the valve gear used rather than because of size. Some engines using valves of

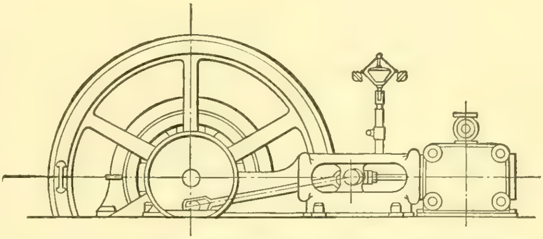

Fig. 365.—Low-Speed Engine with Corliss Valve Gear. Direct-Connected to an Electric Generator.

the poppet, gridiron, and other types, also belong to the low-speed class.

(f) The **uniflow engine,** Fig. 366, may be designed to give thermal economies as good as, or slightly better than, those obtained with other types, when operating *non-condensing*; but it is inherently best adapted to *condensing operation* and to the use of relatively high steam pressure and temperatures. Under these latter conditions, the uniflow may be expected to give a thermal efficiency which is better than that of the single cylinder counterflow, and which compares favorably with that of the compound- or even the triple-expansion counterflow engine. For a given piston speed and power output, because of the lower mean effective pressure, the piston will need to be of larger diameter in the uniflow than in the counterflow. Also, since the uniflow piston has a length of about nine-tenths of the stroke, the cylinders are comparatively long. In consequence it may be expected that the uniflow will cost more than other types of single-cylinder engines, but it will be slightly less expensive than a multiple-expansion engine. Since poppet or piston valves are commonly used to control admissions and cut-off, the uniflow is suitable for medium- or high-speed operation. The commercial sizes usually range from 50 to 1500 hp. A uniflow engine which is properly proportioned to operate condensing would have an excessive compression if changed to non-condensing. As large clearances are detrimental to steam economy, on engines which are to operate normally without vacuum it is usual to have an *auxiliary exhaust valve* (or a transfer valve) to give a *late* compression, the main discharge at release, however, being through the central ports so as to maintain as much of the uniflow principle as possible.

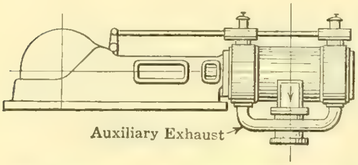

Fig. 366.—Small Uniflow Engine.

On a condensing uniflow engine which may lose its vacuum in an emergency, or which for other reasons may be operated non-condensing at times, it is customary either to incorporate extra clearance space which may be opened to the cylinder by hand or automatically, or to provide auxiliary exhaust valves, resembling those just mentioned, which are brought into or out of action either manually or by the change in back pressure.

(g) The foregoing types constitute the more common of the *commercial* forms of *stationary* engines, but they are, in general, also illustrative of other types. Steam engines that are constructed in combination with pumps, air compressors, ammonia compressors (for refrigeration), hoisting apparatus, tractors, locomotives, etc., are in most respects similar to those considered, but have such modifications as may be necessary to adapt them to their special purposes. Multi-cylinder simple engines and multiple-expansion units also have parts similar to those on simple engines, being merely combinations of several cylinders or engines with such structural modifications and interconnections of piping and steam passages as may be necessary. Multiple-expansion engines may, of course, be arranged for bleeding from between cylinders for heating or for use with the regenerative, reheating, or regenerative-reheating cycles.

The sizes and operating conditions mentioned in connection with the discussion of the various commercial types represent the more usual practice and in many cases are departed from widely. For example, a four-cylinder uniflow engine of 30,000 hp. capacity, with a piston speed of 1500 ft. per minute, has been built [3] and there are a great many examples of other very large or small or special engines.

(**h**) As the different types of *governors*, *valves*, and *valve gears* used have an important influence on the design, performance, and selection of engines they will be considered more extensively in Chapters XXIV and XXV.

286. The Prime Mover in Transportation.—(**a**) The modern world is almost entirely dependent upon some form of heat engine for the transportation of its food, fuel, raw materials, manufactured articles, and passengers. Therefore this field is one of the utmost importance as well as one of intense interest to the engineer because of the variety of problems involved. Each method of transportation may have one type of prime mover that is far superior to any other, or several types may be about equally satisfactory, and the features that make this true will be briefly discussed.

(**b**) *For aeronautical work* the internal combustion motor is exclusively used primarily because of its *light weight* and *fuel economy* which make long, non-stop flights possible. Its compactness combined with the ease of starting and simple control also help to make this type of engine a most satisfactory power plant for this field. *Automobiles*, *trucks*, and *buses* are also nearly all driven by internal combustion engines for the reasons given above and also because of their low first cost for this class of work. In all of these applications, except aeronautical ones, transmission devices are essential to provide the necessary change in torque, speed, and direction of rotation between the drive shaft and the engine shaft. Sometimes gears are thus used in aeronautical work.

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[3] "First and Largest Uniflow on Blooming Mill," 36 in. × 60 in. cylinders, Power, June 22, 1926, p. 968. Also "Giant Uniflow Engine," 60 in. × 66 in. cylinders, Power, July 28, 1925, p. 122.

(c) The *transportation of water, oil, and natural gas* is now chiefly dependent upon pipe lines and pumping stations. In these stations both centrifugal and reciprocating pumps are largely used and they are driven by steam turbines, reciprocating steam engines, internal combustion engines, and electric motors. In this work no single type of pump or engine is exclusively used and a very careful study of all local conditions is required in any specific case before the most suitable selection can be made. In general, pumping stations of this kind have almost full load operation all the time, consequently the energy consumption of the plant is always a very important one.

(d) The *railroads* depend primarily upon the reciprocating steam engine as built in the modern *locomotive* for the transportation of most of their freight, less than 5 per cent being hauled otherwise. The Diesel locomotive is used in small powers for such purposes as switching or drawing light passenger trains and is being developed for fairly heavy trains. The electric locomotive is suitable where the traffic, whether freight or passenger, is sufficiently dense to justify the very high first cost involved. The energy supplied to operate the electric locomotive must come from a central station which, with the transmission system, involves heavy first cost. The steam *turbine locomotive* is still in the experimental stage, the chief problem being to bring the cost down to a point where its use is justifiable. To be economical the turbine needs a condenser and the latter involves a large investment in equipment for cooling the circulating water or other medium that may be used in the locomotive condenser. Of course a condenser would also improve the thermal performance of the piston type of locomotive, but not so much as that of the turbine. The turbine cannot be connected directly to the drivers of a locomotive as it has to run at a very much higher rotative speed. The drive is therefore either by gears or electric generator and motors and the latter is very expensive.

(e) The *steam locomotive* has been so much improved since 1900 that its average fuel consumption is now less than half of its former value and its maximum size more than doubled. The chief reasons for the improvement in thermal economy have been due to using superheaters, increased steam pressure, feedwater heaters, mechanical stokers, improved furnaces, better valves, and limited cut-off. Compounding has not been extensively used in America because of the greater maintenance costs; it is not even used in such enormous articulated locomotives as those of the Northern Pacific, built in 1928, in which the steam pressure is 250 lb., the weight of the locomotive is 715,000 lb. and that of the tender 401,000 lb. The tractive force of this locomotive [4] is 139,900 lb. without the booster, which will give 13,400 lb. more.

The *steam pressures* used in American locomotives have been about as follows: [5]

Year	Range of Pressures	Pressure of Majority	Year	Range of Pressures	Pressure of Majority
1880	130 to 160	130	1905	160 to 215	200
1885	130 to 175	130	1910	150 to 215	200
1890	130 to 180	160	1915	170 to 210	180
1895	160 to 190	180	1920	160 to 220	200
1900	160 to 200	190	1925	180 to 225	200

The 1930 maximum standard is 250 lb. The tendency is clearly toward higher pressures and the Delaware and Hudson Co. has been developing a cross-compound type as follows:

1924, The Horatio Allen, 350 lb., 600 deg. fahr., $23\frac{1}{2}''$ and $41'' \times 30''$
1927, The John B. Jervis, 400 lb., 700 deg. fahr., $22\frac{1}{2}''$ and $38'' \times 30''$
1930, The James Archbald, 500 lb., 750 deg. fahr., $20\frac{1}{2}''$ and $35'' \times 32''$

All of these locomotives [6] have boilers made up of water-tube fireboxes and fire-tube barrels.

In Europe high-pressure locomotives are being developed in several different types,[7] which cannot be discussed here. However, the marked savings in fuel and water that have been obtained, as shown by tests on some of these types, are indicated in Table XX taken from Mr. Stuebing's paper.

Every improvement on a locomotive generally increases its *first cost*, an item of the utmost importance in railroad work. This may perhaps be better appreciated by noting that the cost of operation of steam locomotives in the United States is distributed about as follows:

	Per Cent
Fixed charges	20
Labor	18
Maintenance	20
Miscellaneous materials	8
Fuel	34

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[4] See Table by C. B. Peck, Mech. Eng., Apr., 1930.

[5] Mech. Eng., Nov., 1929, p. 861.

[6] Railway Mech. Engineering, Feb., 1925, April, 1927, and July, 1930.

[7] "High-Pressure Locomotive Developments," by A. E. Stuebing, Trans. A.S.M.E., Railroad Division, 1929, RR-52-2.

TABLE XX

Comparative Test Records of Low- and High-Pressure Locomotives

Section of Railroad	Load in Tons, Excluding Locomotive	Type of Train	Locomotive Boiler Pressure, Pounds per Square Inch	Coal Consumption		Steam Consumption		Percentage Savings, High Pressure against Low Pressure	
				Pounds Total	Pounds per Brake Hp-Hr.	Pounds Total	Pounds per Brake Hp-Hr.	Coal, Per Cent	Steam, Per Cent
Winterthur-Romanshorn-Winterthur	350	Express	170	2760	4.1	23,600	35.0	35.5	51.0
			850	1890	2.65	12,300	17.2		
Winterthur-Romanshorn-Winterthur	350	Passenger	170	3740	3.64	31,700	30.8	30.5	36.0
			850	2660	2.50	20,500	19.7		
Winterthur-Stein-Sachkingen-Winterthur	445	Direct goods	170	3410	4.08	30,600	36.7	25.5	38.3
			850	2460	3.04	18,500	22.6		
Winterthur-St. Gallen-Winterthur	300	Express	170	2590	4.08	23,100	36.3	38.4	42.5
			850	1730	2.51	14,500	20.9		

These values vary with different roads and various operating conditions, but the fact that the fixed charges and maintenance amount to more than the cost of fuel is extremely important. Since the average revenue per ton-mile of freight hauled by the railroads of the United Sates is less than 1.1 cents and since an average of only about 125 pounds of fuel was required in 1929 to move 1000 tons of freight and equipment, including locomotive and tender, a distance of one mile, the significance of fixed charges becomes apparent. The first cost of a steam locomotive is only about half of that of the Diesel-electric of the same size, but the fuel consumption of the latter may average only about one-fourth as much as the former. However, the cost of a unit weight of the Diesel fuel is much greater than the same weight of coal as used by a locomotive. The time required for maintenance items, such as cleaning boilers due to the use of raw feedwater, is much greater than that required for the oil engine. On the other hand, traffic conditions are often such that locomotives must be idle a considerable portion of the time whether they are ready for use or not. Consequently these facts combined with the developments that are now taking place in both types of locomotives make an interesting study in selecting the most economical locomotive to use in any given case. At the present time there are more than 63,000 steam locomotives in use in this country and only a very few oil-electrics, but the latter have been developed in only a few years.

(f) *Marine prime movers* consist of steam reciprocating engines, steam turbines, Diesels, and Ottos. The Otto engine is used extensively and exclusively on small pleasure boats for the same reasons given in Sect. **286(b)**. For medium-sized vessels the Diesel engine is being used more extensively each year. Thus, for example, in 1929 the number[8] of merchant vessels in the world, excluding those of less than 100 gross tons and all Great Lakes vessels, was as follows:

Vessels with reciprocating steam engines	24,359
Vessels with steam turbines	1,361
Vessels with oil engines	2,058

But the marine engines under construction in the world at the end of June, 1930, were as follows:[9]

Reciprocating steam engines	426,093 hp.
Steam turbines	554,260 hp.
Oil engines	1,430,417 hp.
Total	2,410,770 hp.

For naval work, all of the submarines use oil engines, and all of the battleships use steam turbines or engines.

The application of Diesels to fairly large passenger ships is a very recent one. For example, the *Lafayette*, a 25,600 ton vessel of this type, was put in service early in 1930. This ship is 610 ft. long and has four propellers, each shaft being direct-connected to a 6-cylinder, "2-cycle," double-acting, air-injection, Diesel. The cylinders are 23.62 in. × 35.43 in. and each engine is rated at 4000 b.hp. at a normal speed of 145 r.p.m., with a maximum output of 4500 b.hp. at 150 r.p.m. These engine speeds are very suitable for direct connection to the propeller. The *Lafayette* has a service speed of 17.5 knots and will accommodate more than 1000 passengers. Since the engines are of the 2-cycle type they may be easily reversed and thus full power astern obtained. The advantages of Diesels in marine work lies in their better fuel economy, smaller operating force, and smaller space occupied.

The *steam engine* is extremely well suited for propeller speeds in either direction, but it cannot be built in the enormous sizes that are now needed in our fastest steamers. For the medium sizes there is still a large demand for it, as shown by the construction figures previously given. The piston steam engine is used on about 90 per cent of the world's vessels and is commonly of the vertical triple- or quadruple-expansion type.

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[8] Brassey's Naval and Shipping Manual, 1930.

[9] Engineering, July 25, 1930, p. 114.

The *steam turbine* requires some form of speed reduction between it and the propeller if each is to rotate at its most effective speed, as the propeller cannot be operated at a very high speed without causing cavitation. The advantages of *high-speed turbines* in ship propulsion as compared with low-speed ones are: lighter weight, smaller space, lower first cost, higher thermal efficiency, less likelihood of misalignment, and greater flexibility in maneuvering due to ability to endure sudden stops and starts.

(g) The *relative merits* of *gears* and *electric drives* for steam turbine ships are very much discussed by engineers at the present time. The matter cannot be treated fully here but the chief points will be given as they show something of the special problems and determining features involved in work of this kind. The following points are abstracted from an article on "Modern Turbine Propulsion" by E. W. Thau,[10] as his statement of the case is generally considered a very fair one.

1. The *dependability*, *maintenance*, and *operation* of modern types of gear and electric drive are about *equally satisfactory.* Comparisons of the older type of one drive with the modern one of the other is not fair, as they have both been greatly improved.
2. The *weight* of the modern geared drive is only from 50 to 60 per cent of that of the electric.
3. The *first cost* of the modern geared drive is, for the ordinary type of ship, less than that of the electric, and this differential is increased for large power.
4. The *efficiency* of the gear-drive is superior when considering all the auxiliaries in each system; the percentage difference in fuel rates becomes less for the larger installations.
5. The *operating cost* generally favors the geared turbine, but the difference is negligible in some cases and appreciable in others.
6. The *astern power* of the electric drive is 100 per cent while the astern turbine of the gear drive is generally made to be only from 25 to 70 per cent of the main turbines, depending altogether on the requirements. Ordinary ships are built to use the astern turbines only a very small portion of the time and these small astern turbines may therefore be designed for very large torque, by using a large flow of steam, so that full astern propeller speed may be attained with one type as quickly as with the other. But

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[10] Director of Marine Engineering, Westinghouse Electric and Mfg. Co. See Trans. Soc. of Naval Architects and Marine Engineers, 1929.

for such service as ferry boats, the electric drive is superior in this regard because the efficiency of its drive is the same in each direction.

7. *Racing of the propeller* when partially out of the water can be avoided with both types of drives by a speed governor on the turbine.
8. The *arrangement of the machinery* in the vessel is more flexible with the electric drive. This is an item that may be of considerable importance in some cases such as in naval vessels.
9. The *electric drive* has the following intangible advantages:
    - *a.* Torsional vibrations from the propeller and shafting are not transmitted to the ship structure.
    - *b.* An emergency set-up can be made much quicker in case one of several main turbines is disabled.
    - *c.* Unidirectional turbines are simpler than reversing ones.
    - *d.* Inspection and repair of a main condenser may be made by shutting down its prime mover, the motors meanwhile receiving energy from other units.
    - *e.* Balancing and adjustments of turbine rotors after installation are more conveniently made.

(**h**) Two *illustrations of large marine turbine installations* will be given, one using gear drive and the other electric. The first one is the S. S. *Bremen,*[11] which has a displacement of 51,860 tons, a length of 940 ft., a breadth of 101 ft. $8\frac{1}{2}$ in., and 4 turbines, each connected by single reduction gears to its propeller. Additional details are as follows:

Total shaft horsepower	92,500
Astern turbines	65 per cent
Turbine speed	1800 r.p.m.
Propeller speed	180 r.p.m.
Diameter of propeller	16 ft. $4\frac{7}{8}$ in.
Boiler pressure, gage	338 lb. per sq. in.
Temperature at throttle	662 deg. fahr.
Service speed	27 knots.

The *Bremen* and her sister ship the *Europa* are probably both capable of developing about 160,000 shaft horsepower and thus of crossing the Atlantic at an average speed of about 29 knots, if it ever becomes commercially desirable to use such an enormous increase in power for a small gain in speed.

The second illustration is the U. S. S. *Saratoga,* the Navy's first

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[11] Marine Engineering, July, 1930.

TABLE XXI

TRIAL TRIP DATA OF U. S. S. SARATOGA

Test No.	1	2	3	4	5	6
Average speed, knots	33.91	31.67	26.68	18.9	16.9	10.72
Average shaft horsepower developed	183,655	125,735	71,087	26,495	18,975	4659
Slip of propellers, per cent	16.14	11.31	9.74	10.71	10.27	8.54
Number of main turbines in use	4	4	2	1	1	1
Number of main motors in use	8	8	4	4	4	4
Speed of turbine-generator, r.p.m.	1,668	1,483	1,238	1,752	1,562	980
Speed of motors, r.p.m.	309.24	273.07	226.06	161.86	144.01	89.87
Main generators, volts	5,379	4,636	3,911	5,137.5	4,200	2063
Frequency, cycles per second	57.2	50.1	41.25	59.9	53	33
Efficiency of main motors, per cent	96.36	95.87	95.65	93	92.7	89.9
Average steam pressure, pounds per square inch, gage:						
At boilers	302	290	284.2	276	276	278
At turbine steam chest	264.4	271.4	263.3	264	268	276
In 1st stage	188	118	153	99.4	66.1	9
Average steam temperatures, degrees fahr.:						
At boilers	500.8	483.8	485.4	474.7	475.9	464.6
At turbines	488.3	482.4	484	471	470.2	451.8
Average condenser pressure, in. Hg absolute	2	0.9	1.5	1	1.4	1
Fuel consumption (19,500 B.t.u. per lb.):						
Pounds per shaft hp-hr	0.9497	0.965	1.1128	1.137	1.187	2.3806
Tons per mile	2.296	1.705	1.324	0.712	0.595	0.463

large airplane carrier, which has a length of 888 ft., a beam of 106 ft., and (at the time of her trials) a displacement of 39,000 tons.

The power plant contains 4 main steam turbines, each direct-connected to an alternating current generator rated at 35,200 kw. There are 4 propellers, each 14 ft. $10\frac{5}{8}$ in. in diameter, and each propeller is driven by 2 induction motors mounted on the shaft. The speed of the ship is controlled mainly by raising or lowering the speed of the turbine-generators, thus producing a change of voltage and frequency. The motors are also arranged to have 2 different pole combinations; thus with 44 poles the turbine speed is about 11 times that of the propeller, and with 22 poles the ratio is about 5.5 to 1. Since the trials of this vessel contain so much interesting information a short abstract [12] of them is given in Table XXI. These trials were all of 4 hours' duration and were made Dec. 11 to 18, 1928; after their completion the vessel was run over a measured mile at a speed of 34.99 knots, the power required for this speed being 212,702 shaft horsepower. Note that this power is nearly 16 per cent more than that required for a speed of 33.91 knots, or in other words the power required at these speeds increases about as the 5th power of the speed, whereas at about 25 knots the power required increases more nearly in proportion to the cube of the speed. The table also shows that the power plant gave its best thermal performance when working at high powers, but the amount of fuel required to propel the ship a mile is a minimum at about 10 knots.

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[12] For the complete data see Trans. Soc. of Naval Architects and Marine Engineers, 1929, p. 208–213.

CHAPTER XXIV

GOVERNORS AND GOVERNING

287. Governing.—(a) In general the term "governing" applies to the process of adjusting one or more of a machine's operating characteristics, such as the rate of flow or pressure of fluid discharged from a pump or compressor, the speed of an engine, or the power output and frequency of a turbine-generator, to accord with the requirements of the service to be satisfied.

With some kinds of governing the speed is allowed to vary and with others it is maintained substantially constant. In general, at any instant, the speed of a prime mover is controlled by the relation between the force, F, which the working substance produces to make the engine move, and the different portions of that force that are used to overcome the various resistances encountered. One portion, f, overcomes the friction of the mechanism, another part, w, does useful work, and the remainder, a, if any, accelerates the various moving parts. In all cases $F = (f + w + a)$. In order to have the speed remain constant a must be zero, and consequently F must equal $(f + w)$. If F is greater than $(f + w)$, then a is positive and the engine's speed will increase while this condition holds; and if F is less than $(f + w)$, retardation of speed results. An engine which has been running uniformly will acquire greater speed if F increases or if w is reduced; and it will slow down when the opposite changes occur.

The *rapidity* of the acceleration or retardation of motion is dependent not only on the value of $a = F - (f + w)$ but also on the inertia of the parts moved. Thus the cyclic fluctuations in speed that occur during each revolution of a reciprocating engine, because of the varying turning effort, or those disturbances that result from a *rapidly* fluctuating external load, may be reduced to any desired magnitude by providing a sufficient amount of inertia in the flywheel of the engine or in the other revolving masses, such as the rotor of a generator, that are directly connected to the machine. When the external changes affecting the operation of the prime mover take place so gradually as to extend over several revolutions or cycles (as occurs, for example, when either the steam pressure or the load on a steam engine changes slowly) the flywheel effect will reduce the rate of variation, but the change of speed is cumulative and may become undesirably great or even dangerous unless some form of "governing" is employed. Governing is the process of maintaining the proper relation between F and $(f + w)$ to give the kind of regulation desired.

(**b**) A *prime mover* may be governed in any of four ways: It may be (1) *hand-controlled*, as in the case of automobile, locomotive, hoisting, marine, and some rolling-mill engines; (2) *resistance governed*, its operation being controlled directly by the external resistance against which it works; (3) *automatically regulated by a "governor"*; or (4) controlled by a combination of any of these methods. The first method needs no explanation.

(**c**) **Resistance governing** occurs when the opposition to the motion of the prime mover increases considerably with the speed and depends primarily on that factor. It is exemplified by the method of functioning of a steam-driven, direct-acting *pumping engine* which delivers its fluid into a closed reservoir or system of piping. If the discharge pressure is below that of the engine's capacity to produce, the unit will function to raise this pressure until the limit is reached, when the machine then becomes inoperative. With subsequent withdrawal of fluid from the system, the resulting drop in pressure will cause the engine to start again, and it will continue in action until the limiting pressure is once more restored. In this example it is the pressure-variation of the fluid pumped that controls the operation of the engine and no outside means of regulation is required for normal running. However, it may be necessary to provide the engine with a safety stop or speed-limiting governor, to prevent it from running away should there be a sudden demand of large magnitude for fluid, or should a rupture in the piping or other parts of the discharge system occur.

Another example of resistance governing is given by an engine which is used for *ship propulsion.* Since the power required to overcome a vessel's resistance to motion commonly increases about as the cube of the speed, it follows that for any given adjustment of the engine's valve gear, the vessel, starting from rest, will increase in speed until the resistance reaches the limit of the power output for that valve adjustment, after which the speed will remain constant at the value thus attained. For each different adjustment of the valve gear there will be a corresponding speed of the ship. Although the locomotive and automotive engines are hand controlled, a limiting speed is reached in the same manner for each valve adjustment.

(**d**) **Automatic governing** is generally used to maintain substantially constant some one factor, or regulated quantity. The more common forms of automatic governors may be classified as (1) *pressure-responsive*, (2) *speed-responsive*, and (3) *combination* types. Only the first two of these will be considered here.

288. Pressure-Responsive Governors.—(a) When fluid from a **variable speed** displacement pump or compressor having reciprocating pistons or plungers is to be delivered under substantially constant pressure (and especially when the actuating prime mover is larger than is normally needed, so as to be able to meet emergency conditions rapidly) the operation of the prime mover may be controlled by a "pressure-responsive" governor. In this type of governor the pressure of the fluid pumped acts on a diaphragm or piston, or equivalent, which is balanced by pressure, spring, or a weight, and whose movement regulates in the desired manner the pressure or quantity of working substance supplied to the prime mover. When applied to pumps, these pressure-responsive devices are generally termed "pump governors," and when used on compressors they are called "pressure regulators." Since the pump or compressor operates with variable speed, it may also be necessary to provide it with a speed-limiting governor for safety.

(b) When gas compressors are driven at **constant speed** and are to deliver fluid at a substantially uniform pressure, the operation of the compressor ends of the units may be controlled by regulators of the type that are commonly called *unloaders*. These devices function in various ways. When the desired pressure has been reached, some forms prevent further increase by automatically throttling the intake opening, and others keep the inlet valve either closed or open, vary the clearance space, or stop the compressor. Compressors on which unloaders are used may be either of the reciprocating or rotary types and may be driven by prime movers, electric motors, belts, or other means.

(c) When several pumping units *operate in parallel* to deliver fluid into the same system, they must have a combination of speed- and pressure-responsive governing in order that the work will be properly divided between the machines. The detailed consideration of combination governors and other pressure-regulating devices, and also of governors for constant rates of delivery, is, however, beyond the province of this text.[1] The further discussion of governors in this chapter will be confined to those of the speed-responsive type.

289. Speed-Responsive Governors.—(a) Prime movers for driving electric generators, line shafting, mills, etc., are usually controlled by speed-responsive governors that automatically maintain approximately *constant speeds* of rotation under all loads. The more common types of *Speed-Responsive Governors* may be classified as (1) **Mechanical Governors,** including (*a*) *Spindle Governors* and (*b*) *Shaft Governors,* and (2) **Hydrodynamic Governors.**

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[1] For a more extensive discussion of these governors, see Trink's "Governors and Governing of Prime Movers."

A **spindle governor,** see Figs. 361 and 373, has a rotating shaft (usually vertical and direct-driven by the prime mover controlled) which carries revolving masses, or "weights," the centrifugal action of which is balanced by the forces applied either by gravity, or by springs, or by both, in such manner that these weights have different and definite radial positions for each load and speed of the prime mover. Under the maximum load the weights are at their minimum radial distance from the axis of rotation and are said to be in their inner position, or "in"; and while under minimum load the weights are in their outer position, or "out." The weights usually control the position of a sleeve (or collar) which slides on the spindle and to which is connected the mechanism used for adjusting or operating the valves which control the flow of working substance to the prime mover. Spindle governors are commonly used on slow-speed steam engines, steam turbines, and the larger internal combustion engines.

A **shaft governor**, see Figs. 362 and 390, has spring-balanced weights which are mounted commonly in the flywheel, or in a governor case, on the main shaft of the prime mover, and on steam engines is usually arranged to control the throw, angle of advance, or both, of a valve-operating eccentric in such manner as to effect the kind of regulation desired. High- and medium-speed steam engines usually have governors of this type.

A **hydrodynamic governor,** with changes in speed, varies the pressure of a contained fluid, which in turn is used to adjust the valve mechanism of the prime mover. This type of governor is often used on steam turbines.

(**b**) As speed-responsive governors come into action only as a result of changes in speed, the so-called **constant speed governor** is an anomaly. However, with well-designed and properly-adjusted governors, the amount of variation in speed necessary for their functioning is relatively quite small, hence constant speed or "isochronism" can be approached with them sufficiently close for most practical purposes. With some kinds of governors the speed variation necessary to insure proper regulation is as small as 1 per cent and with others it is 4 per cent or more; but in actual practice, as will be seen later, it is often desirable to use greater fluctuations than these minimum values, even though the governing is nominally for constant speed.

290. Regulation of the Working Substance.—With **steam engines** the adjustment of the power output per cycle is commonly accomplished by changing either the *pressure* or the *volume* of steam introduced into the cylinder at each cycle. With the first method, called **throttle governing,** the cut-off is maintained constant and the power output is altered solely by changing the admission pressure, the *conventional* diagrams for different loads resembling those shown in Fig. 367. With the second method, termed **cut-off governing,** the admission pressure remains constant and the amount of working substance fed to the cylinder is varied, the diagrams being as shown in Fig. 368. The actual indicator diagrams are, of course, quite different from the conventional ones, as was explained in Part I. Although the combination of these two methods is possible, it is seldom used. With **steam turbines** of the full reaction type the power is adjusted by *throttling* the steam. If the first stage of a turbine is an impulse element, the regulation may be accomplished either by throttling or by *varying the number of nozzles in action.* With **internal combustion engines,** as already discussed in Part I, page 220, there are four main methods of regulation used: (1) *quantity governing,* in which the amount of fluid admitted to the cylinder per cycle is varied either by throttling or by the cut-off method, the mixture being of constant richness; (2) *quality governing,* in which the volume of charge is constant, but the amount of fuel in the mixture is varied; (3) *hit-and-miss governing,* in which the number of working cycles per unit of time is varied; and (4) a combination of these methods.

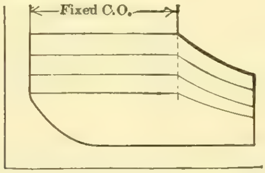

Fig. 367.—Throttle Governing.

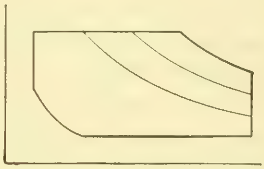

Fig. 368.—Cut-Off Governing.

291. Main Requirements of a "Constant Speed" Governor.—(a) The primary requirements of this type of governor are: (1) *closeness of speed regulation,* i.e., small departure from isochronism; (2) *stability,* or positiveness of position of valves under any load; (3) *sensitiveness,* so that but a small change in speed will cause the governor to shift position and operate the regulating mechanism when a *gradual* change in load occurs; (4) *rapidity* of action under *sudden* load variations; and (5) *governing force* and *work capacity* adequate for operating the control valves and the intermediate mechanism. Some of these factors are inter-related. The methods to be used in testing governors to determine their characteristics of operation are given in the A.S.M.E. Test Code for Speed-Responsive Governors.

(b) The **closeness of speed regulation** maintained by a governor, when the prime mover is operating between the extreme loads A and B, is shown by the **coefficient of regulation,**[2]

$$c = (n_a - n_b) \div \tfrac{1}{2}(n_a + n_b). \quad . \quad . \quad . \quad . \quad (382a)$$

$$= (n_a - n_b) \div n, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (382b)$$

in which n_a = r.p.m. for constant minimum load A (the highest speed;)
n_b = r.p.m. for constant maximum load B (the lowest speed);
and $n = (n_a + n_b) \div 2$ = mean speed.

The factor measures the degree of *total*[3] speed variation under the maximum range of load on the prime mover, and is for the case in which the effect of the friction of the governor parts is eliminated, because the speeds used in the equation are those existing when there is no tendency of the governor to apply force to move the collar. Usually c is given as a percentage of the mean speed. With isochronous governing, if that were possible, its value would be zero.

(c) A governor has **stability** when it has a definite and different configuration of its mechanism, and consequently a definite and different speed, for each load on the prime mover. If the parts of such a governor, while revolving at the proper speed, are purposely displaced from their position of equilibrium, then upon release they will quickly return to that position. With a stable governor, the speed will not fluctuate appreciably while the load is maintained constant at any value within the range of the machine.

If successive loads are applied to a prime mover and the corresponding speeds are determined in each case after the conditions have settled down to constancy, the results may be used for plotting a curve with speeds as ordinates and with loads or collar positions as abscissas. If any portion of this curve is horizontal, it indicates instability over that range of collar movement; for, since there is no definite position of the governor during the corresponding change in load, any disturbance of the equilibrium of the governor will cause irregular speed fluctuations to occur within that range of operation. These fluctuations constitute one form of "hunting." If the whole line is horizontal, it shows that the governor is isochronous and unstable throughout its whole range. If the line has uniform slope the governor has equal stability at all positions. The larger the inclination of the line the greater is the stability, but the poorer is the regulation. Thus stability and closeness of regulation are opposing factors. The more nearly isochronism is approached, the less stable does the governor become; hence there are limits to closeness of regulation that can be used and the value of c adopted in each case depends on the conditions to be met by the prime mover. Since a poorly designed or improperly adjusted governor may be stable in some portions in its range of motion and unstable in other portions, this property should be tested throughout the full range of movement. An unstable governor is often termed an *astatic* one.

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[2] This quantity is termed briefly the "speed regulation" in the A.S.M.E. Test Code for Speed-Responsive Governors. It has also been called the "coefficient of irregularity," the "coefficient of variation," and the "static fluctuation."

[3] For some types of governors it was formerly the custom to give the closeness of regulation in terms of the fluctuation above or below the mean speed, i.e., in terms of one-half of c; hence, when making comparisons of governor performances, it is necessary to have a clear understanding of what is meant by the terms used.

(d) A governor has **sensitiveness** when it requires but an exceedingly small change in speed to cause it to function when the load on the prime mover is altered *gradually.* If the friction and inertia of the regulating mechanism could be eliminated, the Coefficient of Regulation, c, would also measure the sensitiveness of the governor, hence the two terms are often used synonymously. However, if a governor mechanism has internal resistance to change of configuration, the prime mover's load and speed may change a considerable amount before the weights can start to shift position, hence these resistances may have an important influence on the governor's responsiveness to regulation. The A.S.M.E. Test Code for Speed-Responsive Governors therefore specifies that the *sensitiveness* of a governor shall be found by determining the per cent change in speed necessary to cause the adjustment of position of the complete regulating mechanism to take place when the load is altered *slowly.* If a governor is insensitive or sluggish, then, when the power output of the prime mover is being *increased*, the speed existing when passing a given load will be somewhat lower than that at the same load when a gradual *decrease* in output is occurring. Half of the difference between these two speeds can be used for determining the sensitiveness at that load.

For a governor to be sensitive its mechanism must have comparative *freedom from resistance* to movement; hence knife-edge bearings, ball bearings and other frictionless elements are often used in the joints of the mechanism; or the parts may be constantly vibrated slightly, which also minimizes the effect of friction.

(e) The **rapidity** of the action of a governor under an *instantaneous* change in load is measured by the time thereafter required for the prime mover to settle down to constant speed. When such a rapid change occurs, the weights of a governor which has little friction or inertia of parts often first swing past their proper positions and then, pendulum-like, may make several vibrations before becoming stabilized. The maximum change in speed, as well as the time interval, during the period of oscillation should of course not be excessive. This kind of *hunting* may be reduced or eliminated by opposing the motion of the governor by frictional resistance (to the detriment of sensitiveness); hence dashpots (or gagpots), such as will be described in Sect. **292**(e), frictional drags or other damping devices are often added to the governor for this purpose. Other causes of hunting will be considered in Sect. **305.**

(f) The **strength**[4] of a spindle governor for any collar position is the maximum axial force, F, that the collar can exert in that position when friction is eliminated. It is fully developed by slowing down the governor completely from the corresponding operating speed to a dead stop; and its magnitude can be determined readily by measuring or computing the force needed to move the collar back to its operating position when the governor is not rotating. Its value is not necessarily the same throughout the full range of collar movement.

Since but a very small fluctuation in speed is permitted in practice, instead of a complete slowing down, the actual **governing force** that can be utilized in overcoming the resistance of the governor itself and that of the mechanism moved by it, is but a small fraction of the full strength of the governor. Within the usual limits, this force is substantially directly proportional to the percentage of speed-change allowed. The percentage of speed-change to be used to develop governing force depends on the conditions prevailing; for example, in some cases 4 per cent may be allowed, but often either larger or smaller values are selected. For purposes of comparison, the A.S.M.E. Test Code for Speed-Responsive Governors uses the **governing force for 1 per cent change in speed.** With spindle governors, the value of this force is found by actually determining, under specified conditions, the axial load that must be applied to the collar to produce this change in speed. In the case of the shaft governor, the strength and governing force are referred to the eccentric center, or equivalent part moved, and usually their values are more easily determined by calculation based on the governor layout than by actual test.

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[4] This has also been termed less correctly the "power" or "energy" of the governor.

(g) The **work capacity** of a spindle governor is the maximum work that can be done by the collar when it moves through its full range l. To obtain it necessitates a change in speed of 100 per cent. It is equal to $\int Fdl = F_m l$, in which F_m represents the mean strength. Obviously, the *available* work capacity is only a small portion of the maximum. Since the work done at the collar is a direct function of that at the valve, the forces acting at these points are related to each other inversely as the respective distances moved. If the resistance and amplitude of movement of the valve and its mechanism are known, the corresponding work at the collar can be readily found; then, for any designated collar travel, the required governing force follows. It then remains to find a suitable governor that is commercially available, or else design a new one, that will give the desired governing force with the speed variation that is allowable. The A.S.M.E. Test Code provides for the determination of the **work capacity for 1 per cent change in speed.** For shaft governors this factor is referred to the eccentric center, or similar part moved.

In addition to the foregoing main requirements of speed-responsive governors, there are other properties that are of importance, but the discussion of them will be deferred until after some of the fundamental forms of governors have been considered in greater detail.

292. Watt Governor.—(a) A *simple* conical, pendulum, flyball or Watt type of spindle governor, invented by James Watt, is shown in Fig. 369 (a). When stationary, or when operating at its lowest speed, corresponding to the maximum production of power by the prime mover, its position is that shown in full lines, the sliding collar being against the lower stop. Its configuration when at a higher speed, say that corresponding to the smallest power output, is shown by the broken lines. It is a property of this governor that when rotating at any r.p.m., n, within its operating range, the plane of the circular path in which the weights B move is vertically below the pivot O at a distance

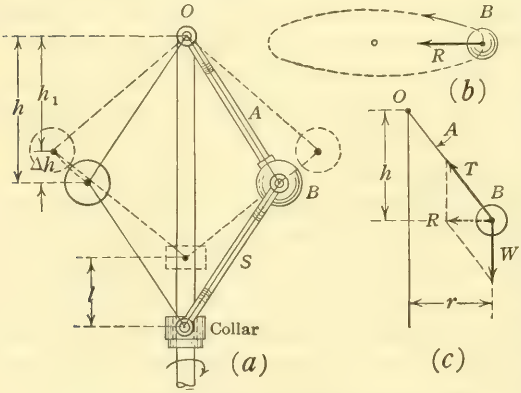

Fig. 369.—Watt Conical Governor.

h which depends solely on n, as will be seen later. This distance h is also the *height of the cone* that is swept out by the arms A and it is usually referred to by that name.

(**b**) When the ball B shown in Fig. 369 (b) moves at a speed of n r.p.m. in a horizontal circular path of radius r inches, there is a resultant force, R, on the ball in the normal direction. The corresponding normal acceleration of the ball is $r\omega^2/12$ ft. per sec.^{2}, where ω is the angular velocity, $2\pi n/60$, in radians per sec. Since the resultant force equals the product of the mass of the ball and its acceleration,

$$R = \frac{W}{g}\frac{r\omega^2}{12} = \frac{Wrn^2}{35{,}200}. \qquad (383)$$

In Fig. 369 (c), the external forces acting on the ball are shown. *Provided the weights of the arms and collar are neglected,* these applied forces are the tension, T, in the upper arm, and the weight, W, of the ball. If the moments of these external forces about the pivot point, O, be equated to the moment of the resultant force, R, about the same point, then,

$$Wr = Rh = \frac{Wrn^2h}{35{,}200},$$

from which

$$h = \frac{35{,}200}{n^2}, \qquad (384)$$

and

$$n = \sqrt{\frac{35{,}200}{h}}. \qquad (385)$$

These equations show that h has a definite and different magnitude for each value of n, as already stated, and that it is independent of the weight of the ball, the length of the arm A, and the method of suspension. Thus, in Fig. 370, the value h measured from the intersections of the arms (produced if necessary) is the same in all the arrangements shown, when the various weights of different magnitudes all revolve at the same speed.

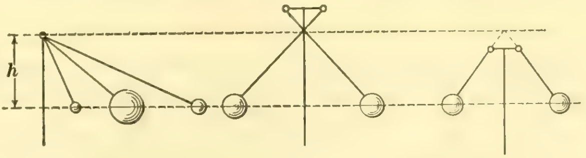

Fig. 370.

Fig. 371 gives the heights of cone for speeds between 60 and 120 r.p.m., advancing by increments of 10 r.p.m. It also shows that the changes in height, Δh_1, Δh_2, etc., corresponding to this increment, decrease as the speed becomes greater.

(c) In order to regulate the flow of working substance to the prime mover, the motion of the collar is transmitted through suitable intermediate mechanism to the valve-controlling mechanism; hence the total collar movement must be made adequate for the purpose. By referring to Fig. 369 (*a*) it may be seen that for a given change in speed the collar lift l is twice the change in height of cone Δh, when the arms A and S are of the same length and are joined together at the center of the weight B. Referring to Fig. 371 it is seen that a given amount of collar movement may be obtained with less speed variation when the r.p.m. is low than when high; therefore these governors are operated at rather low speeds when close regulation of speed, with large collar movement, is necessary. As low speed results in their having little governing force, governors of this type are used only when the parts to be moved are of small weight and are practically frictionless.

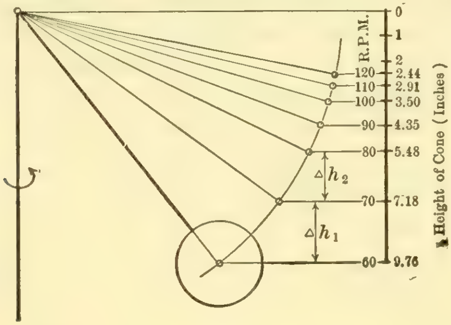

FIG. 371.

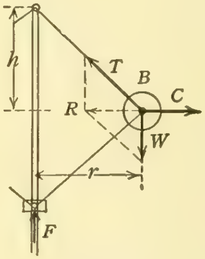

FIG. 372.

(d) As pointed out before, when the weight B, in Fig. 372, is *moving* at some speed n, in a circular path of radius r, there is a resultant force, R, acting on the weight in the normal direction, i.e., *toward* the axis of rotation; and this force, R, is the resultant of two applied forces, T and W. If the mechanism is not rotating, but is *static*, the weight would be in equilibrium in exactly this same position and with the same forces T and W acting, if the force C in the figure is made equal and opposite to R. This force C, which is necessary to hold the weight at any radius r, may easily be measured, as with a spring scale. Thus, it is possible, statically, to determine R, since $R = C$. The corresponding strength F of the governor can be determined by measuring the vertical force that must be applied to the collar to hold it in this static position. If the weight W and the dimensions and position of the mechanism are known, it is possible to compute directly C, R, and F.

Furthermore, since analyses of stationary mechanisms are usually simpler than those of moving ones, governors are commonly treated *statically. This method will be used in discussions that follow.* In other words, the governor weights will be assumed to be in static equilibrium under the action of the *externally applied forces* together with C. Although a static analysis will be made, note that C *changes with speed,* and therefore a separate static analysis must be made for each particular value of speed, and therefore of C required.

(e) Fig. 373 shows a common arrangement of the Watt type of governor, such as is often used for controlling the time of cut-off of Corliss engines. To prevent the weights from moving too rapidly after a sudden change in load occurs, which might cause them to swing beyond their proper position and thus cause hunting to take place, a *dashpot* has been added. This damping device consists of a cylinder containing oil or other viscous liquid, in which is immersed a loosely fitted or perforated piston which is attached to the governor mechanism. If the load changes suddenly the dashpot becomes effective, otherwise it offers little resistance to the movement of the governor. Other forms of damping devices are also used.

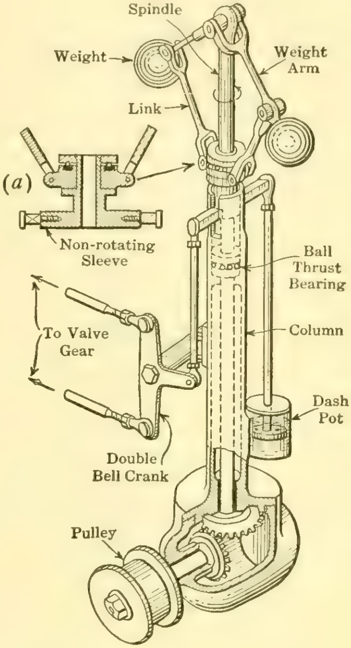

FIG. 373.—Flyball Governor.

293. Porter Governor.—(a) The *weighted* or *loaded* conical governor of the Porter type is shown in Fig. 374 and differs from the Watt governor only because of the addition of the central weight Q which rests on the collar. At any given speed this added weight causes the balls to revolve in a lower plane (i.e., with greater height of cone h) than in the Watt arrangement. Evidently, within reasonable limits, a given value of h can be had at any desired speed by merely placing the right amount of weight at Q.

(b) Each of the suspension rods, *2–3* in this figure, supports its share of the weight Q, the corresponding pull along each rod being S. Taking

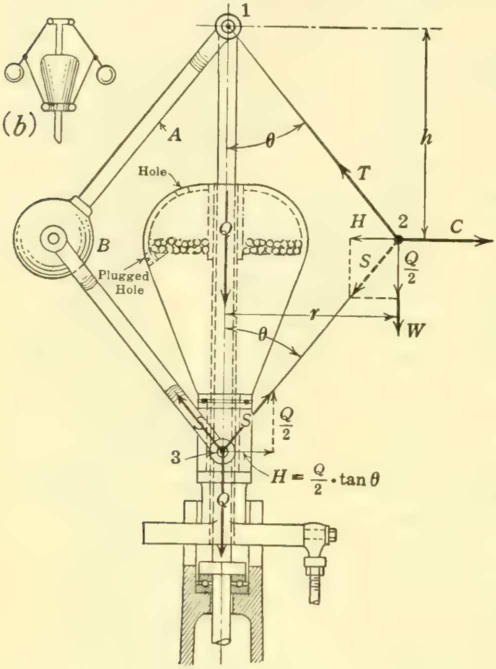

FIG. 374.—Porter or Weighted Governor.

the collar pin *3* as a free body, and assuming that the governor is rhombic in form and has two revolving weights, the vertical component of each force S must be $Q/2$, and the horizontal component is

$$H = \tfrac{1}{2}Q \tan\theta = \tfrac{1}{2}Q(r/h).$$

Using the static treatment, the pin, *2*, at the center of the ball is acted on by the forces S, W, T, and C, the former being resolved into its components $Q/2$ and H in the figure. For any given values of h, r, W and Q, the corresponding magnitude of C required for equilibrium can be readily determined either by an algebraic or graphical summation of the forces, or by equating the moments about *1*. Following the latter method, and remembering that T passes through *1*, it is seen that,

$$\left(C - \frac{Q}{2}\frac{r}{h}\right)h = \left(W + \frac{Q}{2}\right)r. \quad . \quad . \quad . \quad . \quad (386)$$

From which, by substituting the value of $C(=R)$ from Eq. (383) and solving, it is found that the height of cone at any speed n is

$$h = \frac{35{,}200}{n^2}\left(1 + \frac{Q}{W}\right), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (387)$$

and the speed corresponding to any height of cone is [5]

$$n = \sqrt{\frac{35{,}200}{h}\left(1 + \frac{Q}{W}\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (388)$$

---

[5] Note that these equations apply only when the linkage is of the simple rhombic form. For other arrangements, such as that shown at (*b*) in Fig. 374, the equations must be modified.

(c) It is seen that Eqs. (387) and (388) are the same as Eqs. (384) and 385) for the Watt governor except for the multiplying factor in the parenthesis. Evidently the addition of the central weight Q enables the loaded governor to function at a higher speed and at the same time have greater height of cone, more collar movement with the same degree of regulation, and greater governing force, than is possible with the simple Watt arrangement.

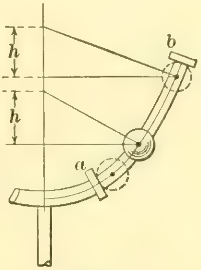

FIG. 375.

294. Isochronous Conical Governors.—(a) Eqs. (384) and (387) show that to have isochronous governing (i.e., $n =$ const.) the height of cone h must always have the same value. Thus in Fig. 375 the path, *ab*, of the ball must be such that the sub-normal to it is constant. As this is the property of the parabola, the ball should be guided over such a path for this kind of governing. Then, with the ball at *any* point on the guide, the forces would always be in equilibrium at *the given speed.*

Such an arrangement is, however, of no commercial value because (1) if a disturbance increased the speed slightly the equilibrium would be destroyed, because the force C would predominate, and the ball would quickly move out to the extreme position against the stop b; (2) a decrease in speed would cause the weight to move to the inner stop a; and (3) the ball has no definite place on its path at the speed of isochronism since it is balanced at any position on the guide at that speed. Hence, while an isochronous governor is ideal as regards constancy of speed, it is *unstable* and merely serves as a limit which actual governors may be made to approach as closely as is possible without introducing uncertainty of action.

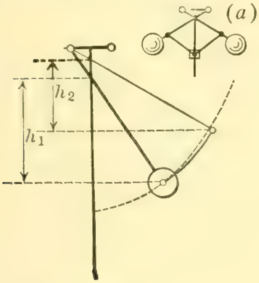

FIG. 376.

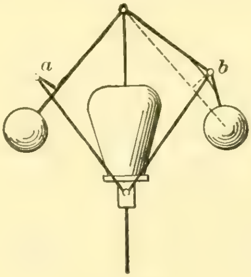

FIG. 377.

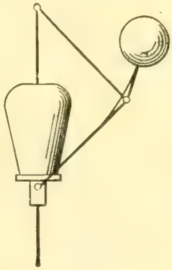
FIG. 378.

With the governors shown in Figs. 369, 373, and 374, which have the suspension points located on the spindle, the path of the ball departs quite widely from the parabola, hence such governors do not give close regulation, except when the collar movement is small, but they are relatively stable.

(b) Fig. 376 shows an arrangement in which the center of the ball moves in a circular arc that approximates a parabolic path, yet departs therefrom somewhat in order that the difference between h_2 and h_1 will be sufficient to insure stability. The complete arrangement is shown at (*a*) in the figure, and is described as a governor having *crossed arms.*

(c) Isochronism can be approached also by having the point of attachment of the suspension arm *offset* from the weight arm, as at *a* in Fig. 377, or by using the equivalent *bent arm* as at *b*. Fig. 378 shows still another arrangement that approximates isochronism. This latter form, known as the Proell, has an *inverted* weight arm. The theory of these types will not be included here.

295. Characteristic Curves.—(a) Given a conical governor with known weights W and Q, then the value of the force C which will effect a static balance with the weight at any radial distance r, with corresponding height of cone h, can be obtained from Eq. (386). If the results of such determination for different values of r are represented diagrammatically with C as ordinate and r as abscissa, as at (*A*) in Fig. 379, a *Characteristic Curve,* or *C-curve* will be obtained. Such curves are useful in analyzing governor performances, since each different arrangement has a distinctive curve.

(b) Referring to Eq. (386), as W and Q are fixed quantities, it is seen that with a given value of the speed n, C is directly proportional to r. Thus for a speed equal to n_1, $C = (Wn_1^2 \div 35{,}200) \times r$, the parenthetical quantity having constant value for that speed. This relationship for $n_1 = 190$ r.p.m. is shown by the straight line on_1 in (*B*) of Fig. 379. For another speed n_2 (say 200 r.p.m.) the corresponding "*n*-line" is on_2. Obviously any force such as C_a will be balanced at the radius r_a, when the speed is n_a, corresponding to the line on_a passing through a.

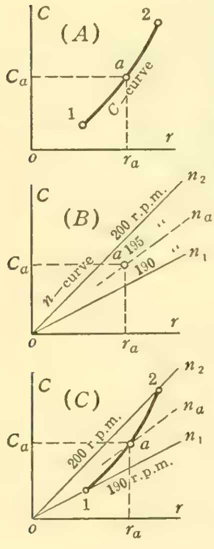

FIG. 379.

(c) If parts (*A*) and (*B*) of the figure are combined, as at (C), it is obvious that the *n*-lines passing through end-points *1* and *2* of the characteristic curve will determine the lower and upper governor speeds, n_1 and n_2, respectively, from which the degree of regulation can then be determined. Note, however, that the *n*-line for the mean speed is not the bisector between on_1 and on_2 since the ordinates of C vary as n^2, and not directly as n.

(d) Should the characteristic curve *1–2* be straight and radial from *o*, it will coincide with some *n*-line and consequently the governor will be *isochronous* at that speed *n*. Evidently such a characteristic curve should be avoided, but should be approached as closely as possible without having instability introduced.

(e) If the *C*-curve *crosses* all the *n*-curves that can be drawn within its limits, as in Fig. 379 (c), the governor will be *stable* throughout its whole range, since for each speed there is a definite radial position at which the forces are completely balanced. The magnitude of the stability at any position is dependent on the angle between the *C*-curve and the *n*-curve at that point.

(f) Should the *C*-curve be like that in Fig. 380, there will be stability for radial positions between points *1* and *a*, and between *b* and *2*. Between *a* and *b*, i.e., at speed n_a, there will be instability, and hunting will occur within this range.

(g) Obviously, by confining the range of action to a shorter portion of a given *C*-curve the degree of regulation can be improved, but there are practical limitations to the smallness of movement that can be permitted in adjusting valve-control mechanisms.

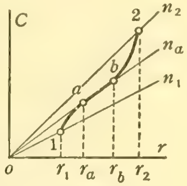

FIG. 380.

296. The Effect of Collar Friction.—(a) For convenience the term "collar friction" is used not only as applying to the frictional resistance of that element alone, but is also extended to include an imaginary force on the collar that is equivalent to the resistances offered by the other parts of the governor and by all the mechanism moved by it. These resistances include those due to friction, gravity, and inertia.

(b) Eqs. (387) and (388) apply only in the ideal case in which there is no resistance to be overcome. If collar friction is present, the speed of the governor must change a considerable amount, Δn, before the force *C* becomes altered sufficiently to overcome this resistance and cause the collar to start to move. Thus, if the change in speed necessary to overcome the resistance when the weights tend to move out equals that when the tendency is inward, there can be a change in speed equal to $2\Delta n$ without movement of the collar. Considering the governor alone, the detention due to friction may cause a variation in speed with collar motionless, from a fraction of 1 per cent to more than 4 per cent, the amount depending on the type of governor, the kind of bearings, the character of lubrication, and whether or not the mechanism is vibrated.

(c) Referring to Fig. 381, if the weight is at any radius r_1, say that for the maximum load on the prime mover, the change in *C* corresponding to *2*Δn is shown by the distance *ad*; and when the weight is at the radius r_2, for minimum load, *2*Δn is represented by the distance *bc*, line *1–2* being the characteristic curve when there is no friction. Starting at *1* with uniform operation, a decrease in load would require an increase *1a* in C before the collar would start to move; then the line *ab* would represent the values of C during the adjustment from inner to outer position of governor weight. If, when *b* is reached, the load is at its minimum value and remains constant, a drop in C will occur from *b* to *2* because under these conditions there is neither upward nor downward tendency of motion of the collar and consequently the influence of friction is nil. If an increasing load is now applied, C will drop from *2* to *c* before any collar movement will occur, and will then follow along *cd*. If when the maximum load is reached, at *d*, it remains steady, point *1* will then show the magnitude of the force C and will then correspond to frictionless operation. Thus, when performing the complete movement of the weight from inner to outer position and return, the values of C vary as shown by the loop *1ab2cd1*. If the *cn*-curves are drawn through the extreme points, it is seen that the corresponding speed range is $(n_b - n_d)$, as compared with $(n_2 - n_1)$ without collar friction. Assuming that the collar friction remains constant throughout the whole range of movement, the Coefficient of Regulation corresponding to the loop is

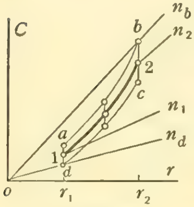

FIG. 381.

$$c' = \frac{n_b - n_d}{n} = \frac{(n_2 + \Delta n) - (n_1 - \Delta n)}{n} = \frac{n_2 - n_1}{n} + \frac{2\Delta n}{n}. \quad . \quad (389a)$$

$$= c + f = c + (f_g + f_m), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (389b)$$

in which c is the Coefficient of Regulation (without friction) and f is the coefficient of insensitiveness, of which the part f_g is due to the internal resistance within the governor and the part f_m is caused by the other mechanism moved. Obviously, if close regulation is desired, all of the various forms of resistance that are included in the term " collar friction " must be reduced to a minimum.

297. Spring-Balanced Spindle Governors.—Some governors use springs to constrain the motion of the rotating weights and in that case the axis of rotation does not have to be vertical, although it is usually so placed for convenience. Several of the many arrangements of spring-balanced governors of the spindle type are illustrated in Figs. 382, 383, and 384. The degree of regulation of the governors shown in Figs. 382 and 383 can be adjusted by changing the initial deflection of the springs by means of the nuts *N*. Such governors can be "set" to give close regulation; and, as they can be operated at high speeds, they can be made to have greater governing force, within the same space limitations, than the types so far considered. Fig. 384 shows somewhat diagrammatically the elements of the Pickering arrangement of governor which is free from articulating joints, the weights being fastened directly to rigidly attached leaf springs *S* which replace the arms ordinarily used. The speed regulation is adjusted by changing the torsion in the supplementary spring *S′*, which by means of arm *C*, shown at (*b*), holds the valve stem collar *A* against the end of the governor rod. The mean operating speed *n* can be adjusted over a wide range by means of hand wheel *B*. In addition to the governor proper, Fig. 384 also shows two safety devices for closing the valve in emergencies. One is operated by the idler pulley *I* that normally rests on the horizontal driving belt. If the belt breaks, or comes off of its pulley, this idler then drops, trips the pawl, and releases spring *S′*, and thus allows the throttle valve to close and stop the engine. Without such provision the valve would be wide open when the governor stops revolving. The second arrangement is hand operated and provides for closing the valve from a distance by means of a cord or rod

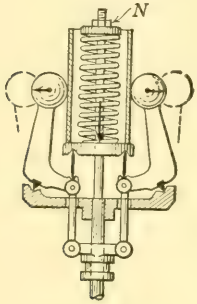

Fig. 382.

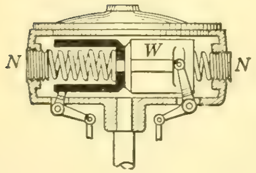

Fig. 383.

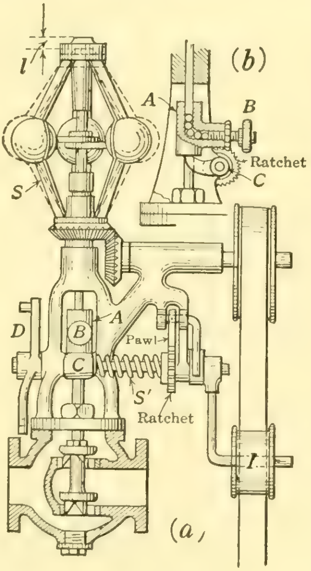

Fig. 384.

attached to lever D. Other spindle governors are often equipped with similar emergency devices.

In principle, the spring-balanced spindle governor is substantially the same as that of the "shaft" type of governor which will be discussed in the next sections.

298. Elementary Shaft Governors.—(a) The simplest elements of a shaft governor are shown in Fig. 385, with the omission of the mechanism controlled. The weight W moves radially along the guides between stops a and b.

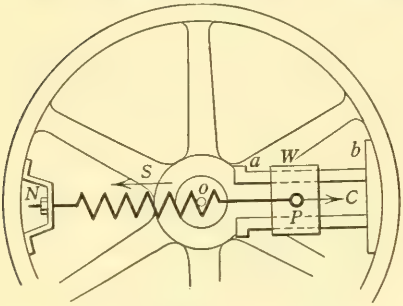

Fig. 385.

For static equilibrium, the force, C, is opposed by the pull S of the spring, the tension of which can be adjusted by nut N.

(b) The pull S can be computed from the total elongation δ and unit strength k of the spring. As this pull is balanced by the force C, it follows that

$$C = S = k\delta. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (390)$$

After having obtained the value of C in any given case, the weight W required to produce this force, when at the corresponding radius r and speed n is, from Eq. 383,

$$W = \frac{35{,}200\,C}{rn^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (391)$$

Or if the weight is assumed, or given initially, the balancing spring pull can be found by reversing this process and determining $C(= S)$.

(c) With the spring so adjusted that it would be just free from tension if the center of the weight W, in Fig. 385, were moved to the center of rotation o, then when the weight has a radial displacement of r, the elongation δ of the spring will equal r, and $C = kr$. The latter is the equation of a radial straight line, such as os in Fig. 386, in which the coordinates are C and r, as in Fig. 379. Obviously this line is a *Characteristic Curve* for constant speed, as it coincides with the n-curve *on*. Evidently, then, for *isochronous governing* δ must always equal r; and if r_1 is the radius to center of W, when the latter is against the inner stop, the corresponding elongation of the spring, called the *initial elon-*

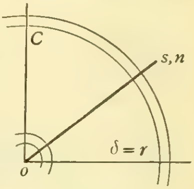

Fig. 386.

gation, is $\delta_1 = r_1$. To obtain this adjustment the nut N in Fig. 385 is turned until the spring is stretched this amount when the weight is against the stop a. But, as has been seen before, the strict isochronous condition must be avoided in actual governors; it is, however, the limit which should be approached as closely as is possible without introducing instability of action.

(d) For *stable governing,* as was explained in Sect. **295** (**e**), the C-curve must be of such character that all n-curves within its range cross it at an angle, none being coincident with it. Thus the curve *1–2* in Fig. 387 would give stability, since W occupies definite positions (such as a, x, and b) for each corresponding speed (n_1, n_x, and n_2). This curve when produced has its origin at o', not coincident with o; and δ is *smaller* than r_1. *Obviously, the condition for stability is that* δ_1 *be made less than* r_1; but the nearer it approaches that value, the closer will be the regulation.

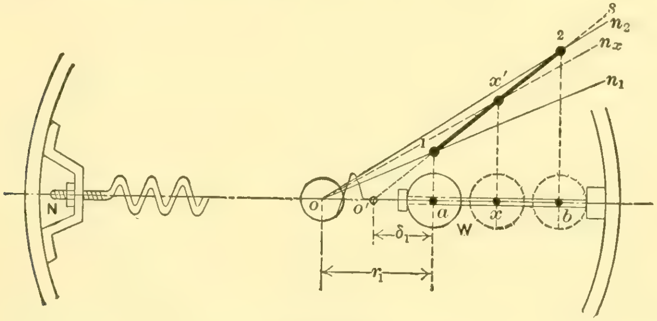

FIG. 387.

(**e**) **The adjusting of a governor of this type,** with a given spring, is divided into two parts: First, the initial elongation of the spring is increased (by turning nut N in the figure) until the best degree of regulation that is consistent with stability is obtained. (Note that the degree of regulation is dependent only on the amount of initial elongation, and that it is independent of the strength of the spring and of the weight of the ball.) Second, the desired mean speed is then obtained by altering the magnitude of the weight of W. If this weight is reduced, the speed will be increased; and if it is made larger, the speed will be decreased.

Should it be necessary to keep the weight unchanged, the speed can be altered by substituting for the original spring one of different unit strength. For the same degree of regulation, however, the new spring would have the same initial elongation as the one it replaces.

(**f**) Instead of being like the elementary form just considered, most shaft governors have the weight W mounted on a "weight arm" to which the spring is attached at a point near the pivot. Fig. 388 shows the simplest arrangement of this kind, with spring at S'. If it is assumed that the arc ab in this figure is the same as the path ab in Fig. 387, the spring S, shown dotted, might be used as in the latter figure. But with the spring actually located at S', its unit strength k' must be greater than that of spring S in the inverse ratio of the leverages, i.e., $k' = k \times (l/l')$, and its initial elongation will be reduced to $\delta'_1 = \delta_1(l'/l)$. The adjustment of such governors is accomplished exactly as described in (e) of this section.

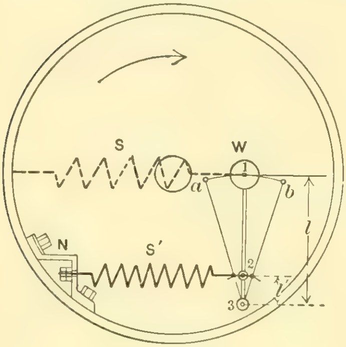

FIG. 388.

299. Analysis of the Usual Arrangement of Shaft Governors.[6]—(a) In commercial governors the ratio of the leverages, l'/l, does not usually remain constant as it substantially does in Fig. 388, and hence the treatment in Sect. **298** (f) does not apply accurately to them.

In the more usual arrangement shown in Fig. 389, x_1 and l_1 are the perpendicular distances from the pivot P to the respective forces S_1 and C_1 for position *1* of the arm. Evidently, for static equilibrium, the spring moment S_1x_1 must balance the moment C_1l_1; hence,

$$C_1 = S_1(x_1/l_1) = \delta_1 k(x_1/l_1). \quad (392)$$

With l_1, x_1, and δ_1 scaled from the drawing, or otherwise determined, and k given, or assumed, C_1 can be computed and plotted with respect to r_1. If the values of C and r for other positions of the arm are similarly determined and plotted, a *Characteristic Curve* results for that spring adjustment, and it can be analyzed as in Sect. **295**. Different initial elongations, δ_1,

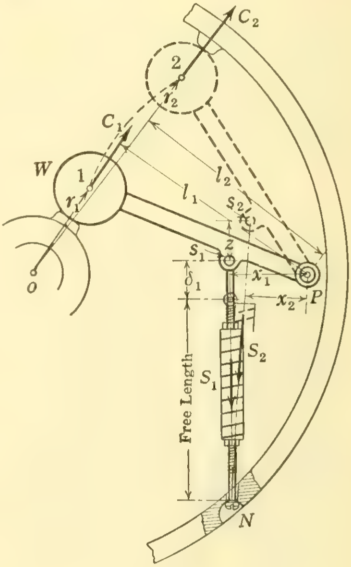

FIG. 389.

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[6] This section may be omitted in short courses.

of the spring will give different curves for any given arrangement of governor mechanism.

(**b**) To determine the spring adjustment that will give a certain desired degree of regulation, the following method can be used: First, referring to Fig. 389, scale the respective values of r, l, and x for inner and outer positions of the arm; and scale z, the increase in length of spring between positions *1* and *2* (i.e., $z = \delta_2 - \delta_1$). Also, for any desired weight W and values of upper and lower speeds n_2 and n_1, compute the corresponding values of C_2 and C_1 from $C = Wrn^2 \div 35{,}200$. Then from Eq. (392):

$$k\delta_1 = (C_1 l_1 \ x_1). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (393)$$

Also

$$k\delta_2 = k\delta_1 + kz = (C_2 l_2 \ x_2). \quad . \quad . \quad . \quad . \quad . \qquad (394)$$

Thus from Eqs. (393) and (394), the required unit strength of spring is

$$k = \frac{(C_2 l_2/x_2) - (C_1 l_1/x_1)}{z}. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (395)$$

With k known, the initial and final elongations are found to be, respectively:

$$\delta_1 = (C_1 l_1/x_1) \div k, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (396)$$

and

$$\delta_2 = (C_2 l_2/x_2) \div k. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (397)$$

Having k, δ_1, and δ_2, and assuming a suitable working stress and radius of coil, the diameter of wire and number of coils in the spring can then be determined. This spring would be adjusted, of course, to have initial elongation equal to δ_1.

(**c**) It can be shown by methods which must be omitted here, that in this case, as in the elementary one, the initial elongation depends solely on the degree of regulation desired and the geometrical arrangement of the governor, and not on the forces acting, i.e., it is independent of W and k. Hence the governor adjustment in this case is accomplished in the manner already given in Sect. **298** (**e**).

(**d**) The results of a simple analysis such as the foregoing may be modified considerably in the actual case by the effects of the gravity, inertia, and frictional forces that act on the parts of the governor and of the valve gear moved. These disturbances are periodic and vary throughout the revolution. The results are also affected by the bowing of the springs and the forces acting on the other revolving elements not included as parts of the primary weights W. However, these secondary influences do not change the method of regulating the governor from that already given. The detailed consideration of these modifying factors is beyond the scope of this text.

300. Commercial Types of Shaft Governors.—(a) In general, the commercial shaft governor for steam engines has one or two pivoted weight arms, on which the action of the centrifugal force of the weight is balanced by one or more springs which are so adjusted that there is a different speed and a corresponding definite and distinct position of the arm or arms for each different load on the engine. The weight arms are connected either directly, or by links, to the eccentric, so that for each speed there is a definite and distinct position of the eccentric, a corresponding cut-off, and a definite amount of power developed by the engine. If the load varies, the speed of the engine will change until a cut-off is automatically found which gives the right amount of power to meet the demand. These governors, if of good design and properly adjusted, may have a coefficient of total speed variation as small as from 1 to $2\frac{1}{2}$ per cent, but often a greater fluctuation is used so as to secure increased stability.

There are two general types of shaft governors, the "Centrifugal" and the "Inertia."

(**b**) The **centrifugal type** of shaft governor may be exemplified by the Sweet governor which is shown in Fig. 390. Pivoted to one of the arms of the flywheel, or governor wheel, is a weight arm which has a heavy head W. When the engine is not running, this weight arm is held in the "inner" position (that shown in full lines) by the leaf spring S. After steam is turned on, the arm will remain in this position until the speed has reached a certain value (for example, 198 r.p.m.), when the force C, exerted on the arm by the weight W, will just balance the spring pull. If the speed is raised further, the increase in this force will cause the spring to deflect and the arm to move outward until at some speed (202 r.p.m.), it reaches the extreme "outer" position (that shown by

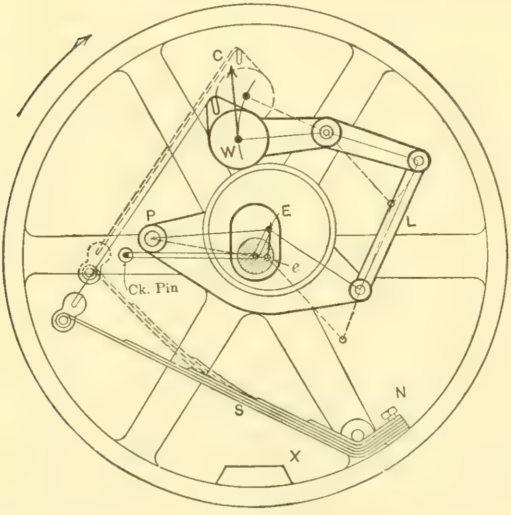

Fig. 390.—Sweet Type of Centrifugal Governor.

the dotted lines) if there is no external demand on the engine. (In the example the mean or normal speed is 200 r.p.m. and the total variation is 2 per cent. By changing the adjustment of the spring the amount of variation can be altered.)

(**c**) Again referring to Fig. 390, it is seen that the arm carrying the eccentric is pivoted at *P* to one of the arms of the wheel, and is connected by a link *L* to an extension of the weight arm. The manner in which the governor operates to control the eccentric is as follows: When the engine is standing still, the eccentric center is at *E*, the position for the latest cut-off. When steam is turned on, the engine will speed up until a certain r.p.m. is reached, at which the governor arm will begin to move out, thus shifting the eccentric center towards *e*, the position for zero cut-off. This movement will continue until a position is reached at which the power developed just equals the load, and as long as the latter is constant the governor arm will remain in this position. Now, if the load is reduced, the engine will speed up slightly and this causes the weight to move out, shifting the eccentric nearer to *e* and reducing the output until it becomes again equal to the demand. Similarly, if the load is increased, the speed of the engine becomes less, and, as the weight arm moves in, the cut-off will be increased until a balance is again effected between the output and the load.

(**d**) The weights and movements of the eccentric and arm on this governor are made such that the center of gravity of their combined masses always remains substantially in a fixed position with respect to the wheel so that the balance is not seriously disturbed by changes in the configuration of the governor parts. The weight *X* on the rim of the wheel in the figure is added to effect both a gravity balance and a centrifugal balance of the wheel and governor. Sometimes a dashpot is used on this and other forms of shaft governors. Its cylinder is usually attached to the rim or an arm of the wheel and its rod is connected to a pin on some governor part, such as the weight arm or the eccentric arm.

(**e**) Fig. 391 shows another arrangement of centrifugal shaft governor; but in this case there are two weight arms, symmetrically placed, instead of one. In its action, this governor is identical with that which has just been described.

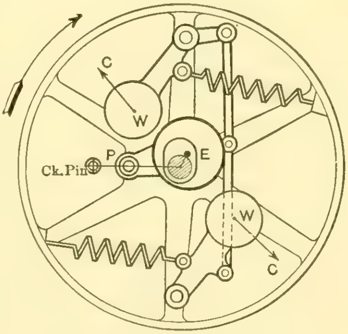

Fig. 391.—Centrifugal Governor.

Centrifugal governors are comparatively large and have heavy weights that are located relatively far from their axes of revolution; in consequence, they have greater governing force and more resistability against outside disturbances than any of the other types.

(f) Fig. 392 illustrates the Rites governor of the **Inertia Type,** which is also shown in Fig. 362. It consists primarily of a long weight arm WW', pivoted at P, and constrained by a spring S. The arm carries the eccentric pin E and its end W' is heavier than W, so that its center of gravity is at G. The "in" position of the parts is shown in full lines with eccentric center at E. This is the position for the latest cut-off, and is also the one occupied when the engine is not running. The "out" position is shown by the broken lines, with eccentric center at e, and is for the earliest cut-off. The direction of rotation is clockwise, as shown by the arrow, and cannot be reversed without making changes in the governor, because the inertia would act in the wrong direction.

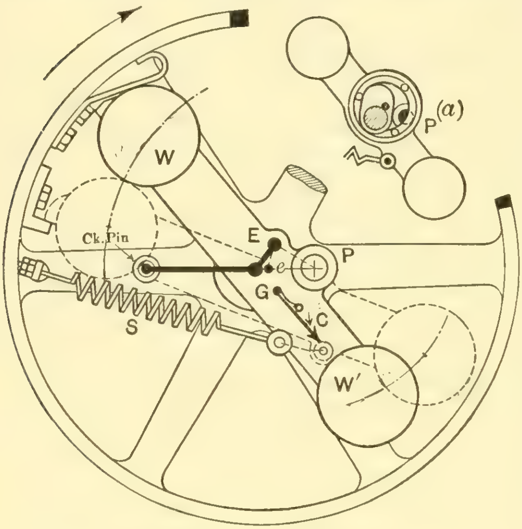

Fig. 392.—Inertia Governor.

When starting up the engine, a certain speed is reached at which the weight arm will leave contact with its inner stop; and in that position, static analysis shows that, taking moments about the pivot P, the counter-clockwise moment of the force C acting through the center of gravity G, is balanced by the clockwise moment of the spring pull. A further increase in the engine speed increases C and its moment, and to effect a balance the spring pull and its moment must also become larger, the spring being elongated. Thus the entire weight arm is automatically moved counter-clockwise about P, and the eccentric center is shifted from E toward e until a sufficiently early cut-off is obtained to balance the load. If the speed is decreased, due to added load, the weight arm will automatically shift clockwise about P and will make the cut-off occur later. In this phase of its operation, this governor acts in the same way as those of the centrifugal type previously studied. However, in addition to this kind of action, there is also an *inertia effect* which assists the movement of the weight arm about the pivot P in the following manner: If the engine speeds up *suddenly*, the weight arm tends to continue to revolve at its now slower speed, because of its inertia, and hence lags behind the wheel, moving counter-clockwise with respect to the pivot P. It is seen that this movement is in the same direction as that caused by the corresponding change in the force C. Again, if the load is suddenly increased, the engine will slow down slightly, but the weight arm will tend to continue at its now higher speed and this will shift the arm clockwise with respect to P, thereby again assisting the centrifugal force in changing the position of the arm. It is seen that the inertia governor is *primarily a centrifugal governor*, but that, in addition, the weight arm is so pivoted, and has its weight so distributed, that its inertia assists in making the adjustment, and that the more *sudden* the change in the load the greater will be the assistance thus rendered.

In order to have the inertia of the weight arm act in the right direction, the *fulcrum pin must be placed on the side of the shaft opposite to the crank pin* when the usual "external" valve is used, and on the same side when the valve is "internal" (Sect. **308**). If the shaft extends beyond the governor wheel, both the governor arm and the eccentric must be made to surround and clear it in the manner shown at (a) in Fig. 392. As the center of gravity, G, of the weight arm is near the center of rotation, governors of this form have relatively less governing force than do centrifugal governors, and hence they should be used only when the parts moved have little friction and small inertia.

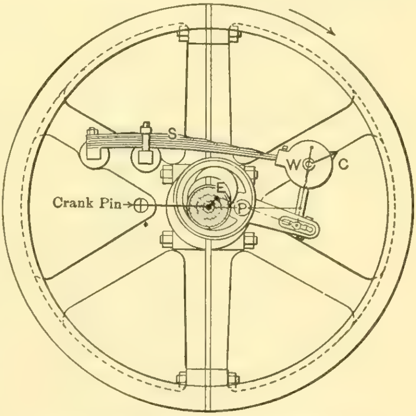

FIG. 393.—Armstrong Governor.

Another example of this type of governor is the Armstrong governor, shown in Fig. 393. This governor consists in its elements merely of the weight W and the laminated spring S. In this device, as in the Rites, the inertia effect assists in producing a rapid adjustment of the engine output when the change in load is sudden.

There are almost an unlimited number of forms of shaft governors, but most of them are merely modifications or combinations of those which have been described. They are all adjusted in the manner outlined in Sect. **298** (e).

301. Hydrodynamic Governors.—Some speed regulators, which may be called **fluid-pressure or hydrodynamic governors,** make use of the hydraulic principle that the pressure head, z, developed by a centrifugal pump varies as the square of the velocity of the impeller tip (i.e., $z = v^2/2g$). Their actuating force, the pressure, therefore varies as n^2, just as is the case with the other speed-responsive governors already considered. This system of governing, which has recently been applied to steam turbines, uses the combination of a small centrifugal pump, mounted directly on the shaft of the prime mover (as in Fig. 308), and a pressure-actuated element (such as a spring-balanced piston) which in turn controls the valve gear either directly, or through a relay system such as will be described in the next section. For each load on the prime mover there is a definite fluid pressure developed within the system, and a definite position of the control piston or its equivalent. Air, water, oil, or any other suitable fluid can be used as the pressure-varying and transmitting medium, and the two main elements of the system may be widely separated without complication, since they are merely connected by simple piping. When oil is adopted as the working fluid, in which case the device is usually termed an *oil governor*, the pump element can also be made to supply lubricant to the bearings of the prime mover, and further, it can be used to furnish high-pressure oil to operate the relay system, if one is used.

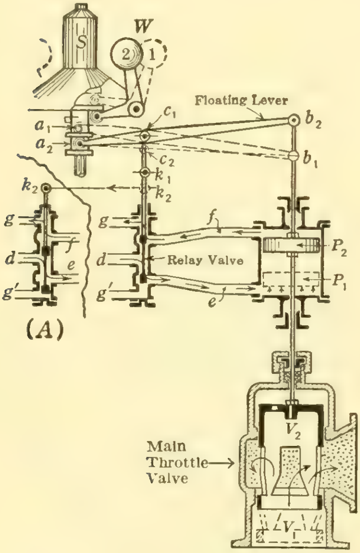

Fig. 394.—Relay Mechanism.

302. Relays—(a) When the valve gear to be controlled offers relatively large resistance to motion, and when it is to be manipulated by hand or by a comparatively weak governor, a powerful *relaying device*, or *servomotor*, may be interposed between the regulator and the gear. The usual relay system has a small, balanced *pilot valve*, or *relay valve*, which can be easily moved by hand or by the governor, as the case may be, and this valve controls the flow of high-pressure fluid to a relatively large piston or diaphragm that operates the control valve, or valves, on the prime mover.

(b) One arrangement of a **relay governor** is shown diagrammatically in Fig. 394 and it functions as follows: Suppose the mechanism to be initially in position 1, shown in broken lines, corresponding to a small load. The speed reduction accompanying an increased demand for power will cause the governor weight W to assume some new position such as 2, shown in heavy line. This change shifts the left end of the "floating lever" from position a_1 to position a_2 and causes the pin which was originally at c_1 to move to position c_2. This lowers the relay valve from its central position to that shown separately on the left, at (A), and thus allows the relaying fluid from pipe d to flow through passage e to the under side of the control piston, which is now in position P_1, and permits the fluid above the piston to escape through passages f and g. The consequent movement of the piston upward shifts the floating lever from position a_2b_1 to a position a_2b_2 that returns pin c to its original position c_1, with the relay valve in its central or closed position. The piston will then stop in position P_2, which is the one it will always occupy when the governor weight is at 2. Thus, with this relay arrangement, which has a rigid return, there is the same definite position of the valve and same amount of opening for each governor position (or load on the prime mover) that there would be if the governor moved the piston and valve directly by means of lever ab pivoted on a fixed fulcrum located at c_1. The relay valve, as shown, is practically without lap (i.e., when centrally placed, its edges do not overlap those of the ports).

(c) A manually operated relay, such as is often used to control the valve gear adjustment on large marine and hoisting steam engines, may be arranged similarly except that the end of the floating lever is controlled by hand instead of by a governor. Often double-beat poppet valves or piston valves are used for the main throttle valve, instead of a sleeve valve, such as that shown in Fig. 394. Sometimes on large impulse steam turbines the relay piston acts through rack and pinion, to turn a shaft on which are cams which open or close in succession a series of valves which control the operation of the nozzles, as in Figs. 298 and 300. For the floating lever, other differential gearing may be substituted for moving the pilot valve, and sometimes double or triple relaying is used. Instead of employing a fluid-actuated piston to move the valve, the operating force may be derived from the prime mover itself through a servomotor, or it may be furnished by an electric motor or solenoid. A relay system relieves the governor proper from disturbances due to the forces which control the valves. The governor used should be as free as possible from internal friction, since its parts are not subject to the vibrating motions caused by the cyclical variations of the prime mover and which minimize the effect of friction.

(d) For hydraulic turbines with long penstocks the relay governor must have great temporary stability during the initial movement. The relay is usually provided with a combination of a spring and oil pot which acts as a compensator with slowly yielding return which gradually restores the speed to its original value. At first the mechanism functions like the relay having a rigid return, but while so doing it compresses the compensator spring which subsequently returns the governor parts to their original positions, thus obtaining ultimate isochronism. There are several arrangements of relay systems of this kind, but they will not be considered further in this text.

303. Speed and Load Adjustments.—(**a**) When a spindle governor of the *gravity-balanced type* is belt-driven, as is usually the case, a permanent alteration in the operating speed of the prime mover can be made by changing the speed ratio between the governor and the engine, generally by changing diameter for one of the belt pulleys; or, if the change is small, it can be accomplished by altering the load on the governor's collar. For the latter purpose the central weight of Porter governors usually contains a pocket, as in Fig. 374, so that shot may be added to increase the equilibrium speed, or be removed to decrease it. Sometimes auxiliary weights or springs are attached to the mechanism so as to change the load on the collar.

(**b**) When the governor is of the *spring-balanced type*, a new normal speed is usually obtained by modifying the revolving weights without changing the springs or their adjustment; or, on some arrangements of shaft governors, it may be effected by changing the leverage of the springs or that of the weights or of both. If the spring arm is altered, a readjustment of the initial elongation of the spring may be necessary in order to maintain the same degree of regulation as before. Changes in the positions of the weights or of the spring attachments on the governor arms may, however, modify the characteristic curve in such manner as to introduce instability. In general, increasing a weight or its moment, or reducing the spring moment, lowers the operating speed; and opposite changes have the reverse effect. These adjustments are permanent and are made when the prime mover is at rest.

(**c**) It is often desirable to make slight changes in the speed of a prime mover while it remains in operation. For example, before an alternating current generator is connected to a distributing line to function in parallel with other units thereon, the speed of its prime mover must be adjusted so as to bring the generator in phase with the others of the system. There are many kinds of **speed-changers** that are used to accomplish this adjustment. Usually these **synchronizers** alter the tension of the main or auxiliary springs on the governor; and this change can be made while the unit is in operation, either directly by hand or by means of a small electric motor that is remotely controlled, as from the switchboard. After the prime mover is on the line, the speed-changer can be used as a **load-adjuster** to make the machine carry any desired portion of the total load on the system, for, at any given synchronizing speed at which all the units must be operating, the spring tension can be altered by it so as to have the governor weights and the valve controlled occupy any desired position in their range of movement. Further, the same devices can be used also as **frequency regulators** for the whole system. Thus, when an increase in load lowers the frequency, the latter can be raised by means of the synchronizer on one of the machines, which will then take more load. The other units now having decreased loads will speed up, thus increasing the frequency of the whole system. As the frequency rises, the spring tension on the first unit must be reduced somewhat to restore the proper division of load. The frequency regulation may be made automatic.

(**d**) When one or more prime movers, group A, operate more economically than the others, group B, that are connected in parallel with them in an alternating current system which has wide fluctuation in load, it is desired to have the A-units develop as much power as possible, i.e., act as *base-load* units, and let the variations be carried by the B-units. This distribution of load can be brought about by so adjusting the governors on the A-units as to cause these machines to be relatively less sensitive than are those on the less economical units. Then, by means of the speed changers, the A-units can be loaded heavily, while the more responsive B-units are lightly loaded except when the power demand is large, thus securing the best results from the combination.

304. Safety Devices.—**(a)** In addition to the main governor on a prime mover it is also advisable to provide the machine with some form of *emergency device* or **safety stop** which will come into action if the main governor fails to keep the speed within safe limits.

When a governor is stationary its mechanism automatically assumes the position for maximum flow of the working substance to the prime mover. Thus, when a rupture of its driving gear stops a governor, the machine which it regulates will run away unless fully loaded. To avoid such a catastrophe, the governing mechanism on *steam engines* is sometimes arranged to have the *valves become inoperative* when the governor weights are fully "in." This is often the arrangement on Corliss engines (Sect. **327** (1)). Then to start the engine it is necessary first to raise the governor collar slightly; usually it is supported on a pawl, or finger, which falls to one side as soon as the lifting speed is passed. A safety device which closes the throttle valve when its belt breaks, or comes off the pulleys, was described in connection with Fig. 384.

Most *steam turbines*, and many other prime movers, are provided with devices called *secondary, emergency, safety,* or **over-speed governors,** which are adjusted to operate emergency trip valves, or other shut-offs, when the speed reaches a value slightly above normal (say 10 per cent greater). There are many other possible arrangements of emergency devices, some of which are automatic and others are hand-operated either directly or by remote control, in the latter case by cords, fluid pressure, or by electric current.[7]

In addition to having speed-limiting devices, large steam turbines are commonly arranged to be automatically stopped when the *oil pressure* in the lubricating system becomes dangerously low, or when the rotor has too much *axial displacement.*

(**b**) When a multi-cylinder steam engine or turbine which has *voluminous intermediate receivers* or piping is subjected suddenly to a large decrease in load, the mass of steam stored between cylinders will still act in the lower stages of the machine, even though the primary governor immediately cuts off completely the main supply of steam, and this stored steam may cause the prime mover to overspeed and possibly be destroyed unless provision is made against such an occurrence. Hence steam engines of this kind usually have the valve gear of all cylinders under governor regulation; and steam turbines are often equipped with over-speed secondary-governors which control suitable valves that in such an emergency will bypass the stored steam so that it will not act on the lower stages.

305. Governor Troubles.—(**a**). **Hunting,** racing, surging, or pumping are the various terms commonly applied to the governor troubles, due to faults in the characteristic curves, to adjustments approaching isochronism too closely, to over-swinging under sudden load changes, and to lack of strength to resist disturbances. The same effect may be brought about also by *faulty adjustments* which allow undue friction, sticking, and sudden release of parts, lost motion, leaks in pilot valves, and other mechanical disturbances; and it may result also when there is a markedly *non-uniform rate of opening* or closing of the control valve with respect to the governor movement.

(**b**) The character of governing, especially with sudden changes in load, may also be dependent on the inertia of the flywheel and other moving parts of the prime mover, which will be called briefly the **flywheel effect.** Thus a heavy flywheel will delay the response of the governor to an alteration in power, whereas one too light may permit the adjustment to occur too rapidly and also may allow cyclic disturbances to take place. The character of governing may also depend on the distribution and amount of **working substance stored** within the machine, for large steam chests and intermediate receivers located beyond the regulating point may contain steam which will continue to act after the main governor has functioned.

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[7] "Standardization of Safety Stops," by W. Greenwood, Power, Oct. 30, 1923, p. 709.

(c) In the **parallel operation** of units delivering alternating current to the same bus-bar, the *relative* rapidities of the speed adjustments of the several prime movers under load changes are often of great importance, because if one unit takes longer than another to accomplish the change when the load varies, its generator may get out of phase with the others sufficiently to result in serious electrical disturbances in the system and may even cause a shut-down. But the rapidity of speed change is dependent both on the action of the governor and on the flywheel effect. Thus one engine with relatively light flywheel will permit its governor to make a rapid adjustment, while another unit with comparatively heavy wheel may have much less responsive regulation, although the two governors themselves may have equal rapidity under like conditions. Consequently, for parallel operation, not only must the governors have suitable characteristics and adjustments, but the moving masses of the units must also be properly proportioned.

REFERENCES

Collins—Shaft Governors.
Tolle—Die Regelung der Kraftmaschinen.
Trinks—Governors, and the Governing of Prime Movers.
Trinks—Shaft Governors.

CHAPTER XXV

VALVE GEARS OF STEAM ENGINES [1]

306. Introduction.—It is assumed that the reader is already familiar with the arrangement and operation of the simple steam engine having the plain slide valve, and that he is able to use at least one kind of "valve-gear diagram" for the analysis or design of a simple D-valve. The purpose of this chapter is to bring out certain useful conceptions and to give a brief discussion of the different types of valve gears used on steam engines. First it is advisable to review some of the definitions and to establish the notation that will be used throughout the discussion.

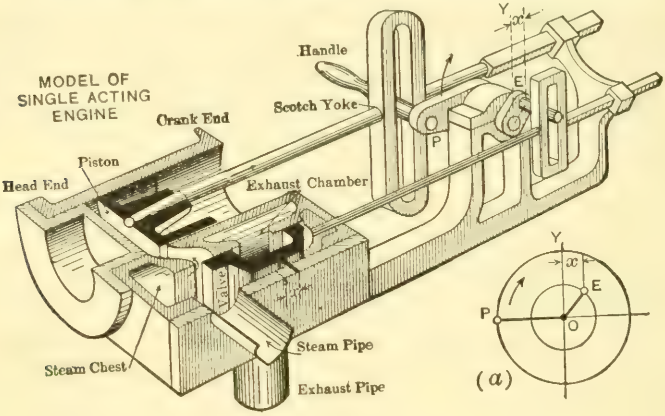

Fig. 395.

307. The Engine. Definitions.—(**a**) The *crank end* (C. E.), or front end, of the cylinder, or valve, is the one nearest the crank, or next to the engine frame. The opposite end is the *head end* (H. E.), or back end.

(**b**) The *forward* (Fd.) *stroke* of the piston, or valve, is that towards the crank. The other stroke is the *return stroke*, or *back* (*Bk.*) *stroke.*

(**c**) Fig. 395 shows a model of a simple single-acting engine with piston and valve driven by crank and eccentric pins operating in *Scotch yokes*, or *slotted crossheads.* It is evident that with such a mechanism both the valve and piston will have simple harmonic motion, and consequently the analysis of the valve action is a simple matter.

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[1] In brief courses the whole of this chapter may be omitted, but it is suggested that at least Sects. **306**, **307**, **308**, **310**, **314**, **317**, **318** and **319**, be studied.

With this arrangement the motions are exactly the same as would occur if the engine had connecting and eccentric *rods of infinite length.*

(**d**) Actual steam engines, of course, have connecting rods and eccentric *rods of finite length*, and the "angularity" of these rods causes the motions of piston and valve to depart slightly from the true harmonic. The eccentric rods are usually so long, however, when compared to the length of the eccentric crank, that the departure in the case of the *valve* is negligible; but the effect of the angularity of the connecting rod on the positions of the *piston* may be of considerable magnitude and usually must be included if accurate results are desired. However, in making generalizations, or in an elementary study, it is often sufficiently exact to consider the motions of both the valve and the piston as truly harmonic, and such an assumption simplifies the investigation greatly, as will be apparent in many of the discussions which follow.

(**e**) The crank is on *dead center* when the piston is at the end of the stroke, and is then horizontal on horizontal engines. When the piston is at the head end of the cylinder, the crank is on the "head-end dead center"; when at the other end, it is on the "crank-end dead center."

(**f**) The *eccentric* (*ecc.* or *E*) is really a crank with pin of such large diameter as to surround the shaft. In the following discussion the term "eccentric" will be used as applying to the center of this pin. Like other cranks, the eccentric has dead-center positions.

(**g**) The *throw* of the eccentric is the "eccentricity" or length of the crank. (There is a lack of agreement in the use of the term "throw," some using it as meaning the total movement of the valve or "travel.")

(**h**) The term *piston displacement*, as used in this Chapter, will mean the distance that the center of the piston has been moved from its middle position, i.e., from mid-stroke.

308. The Valve Definitions.—(**a**) Fig. 396 shows the longitudinal section of a symmetrical D-valve for a double-acting engine. With pure harmonic motion the functioning of the crank end of the valve is the same as that of the head end, except that there is a phase difference of 180°. The following discussion will be confined to the head end, but is of course equally applicable to the crank end. The valve shown is arranged to admit steam to the cylinder past its *outer edge* (i.e., when the *steam edge S* of the valve passes to the right of the steam edge *P* of

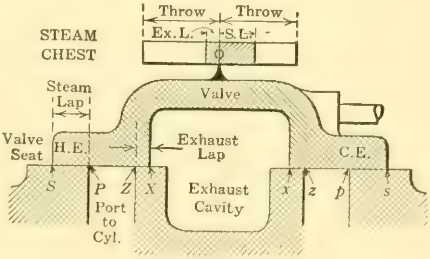

FIG. 396.

the port), and to exhaust the steam from the cylinder past the *inner edge* (i.e., when its *exhaust edge* X moves to the left of the exhaust edge Z of the port).

(**b**) The *width of the port* is the distance PZ in the figure.

(**c**) The valve shown is called an *external valve.* If it were arranged to admit steam to the cylinder past its inner edges and exhaust it at the ends, it would be an *internal valve* (but then would have to be of a design different from that shown). Unless otherwise stated, the valve will be assumed to be external.

(**d**) The terms "*steam chest,*" "*exhaust cavity,*" and "*valve seat*" should not need explanation (see Fig. 396).

(**e**) The *valve is central* (or in *mid-travel*), with index at O, when the eccentric is vertical, either up or down, as will be seen from Fig. 395.

(**f**) The *lap* of the valve is the distance between the valve edge and the port edge with which it operates, when the valve is central. The *outside lap* (or outer lap) is that of the outer edge, and the *inside lap* is the lap of the inner edge of the valve. The *steam lap* (S. L.), see Fig. 396, and the *exhaust lap* (Ex. L.) are, respectively, those of the steam and exhaust edges of the valve. The lap is positive if the port is closed when the valve is central, and negative if open. (Negative lap is sometimes called "clearance.")

(**g**) The *valve opening* is variable and is dependent on the displacement of the valve; but the term is usually understood as referring to the *maximum* width of the opening unless otherwise stated.

(**h**) The *travel* of the valve is the *total stroke* or amplitude of its motion. If the valve is direct-driven, the travel is equal to the diameter of the eccentric circle.

(**i**) The term "*displacement,*" when applied to the valve, will be understood to mean the distance the center of that part has been moved from its central position; and in the case of a horizontal engine it will be the horizontal distance from the eccentric center to the vertical center line of the shaft, as shown by x in Fig. 395 (a).

(**j**) The **four periods of operation** of the valve are admission, expansion, exhaust, and compression.

(**k**) The **four major valve events** are: Admission (A, a), cut-off (C, c), release (R, r), and compression (K, k). (The capital letters given in the parentheses will be used to indicate the respective H.E. events on the diagrams which are to follow, and the small letters will apply to the crank-end events.)

The **four minor events** are: Maximum displacement of the valve to the right (M), same to the left (m), valve central and moving to the left (Q) and central but moving to the right (q).

(1) *Unless it is specifically stated to the contrary, it will always be assumed in the following discussion that the engine is horizontal, with cylinder to the left of the crank shaft, that an external valve is used, and that the crank rotates in a clockwise direction.*

309. Action of the D-Valve and Eccentric.—(**a**) When the valve is driven by a Scotch yoke, it is seen, by referring to Fig. 395, that (1) the valve is central when the eccentric is on the vertical center line OY through the center of the shaft, (2) the valve will be in this position whether the eccentric OE is vertical upward or downward, and (3) that at all times the displacement x of the valve equals the displacement x of the eccentric E. With these relations holding, it is easy to analyze the valve action by using "valve diagrams."

RECTILINEAR DIAGRAM OF VALVE DISPLACEMENTS

(**b**) In Fig. 396 it is seen that the head end of the valve must be displaced to the right a distance equal to the steam lap before opening to steam occurs, and that any further displacement represents opening. In Fig. 397, in which the radius of the circle equals the eccentric throw, the "*steam-lap line*" AC has been drawn at a distance equal to the H.E. steam lap to the right of the vertical axis qQ. Thus, the horizontal distance from the eccentric (anywhere on this circle, for example at B) to qQ is the valve displacement, and its distance to the right of AC is the amount the steam edge is open. Again referring to Fig. 396, it is seen that the valve must be displaced to the left a distance equal to the *exhaust lap* before exhaust opening occurs, and that further displacement in that direction represents the amount of opening. In Fig. 397 the *exhaust-lap line* KR is drawn to the left of qQ at a distance equal to the exhaust lap. Then if the eccentric is to the left of KR, the horizontal distance between it and that line shows the amount the exhaust edge is open at that time. (If the exhaust lap is negative, KR will be at the right of qQ.) The valve will be displaced to the right of its central position while the eccentric is located on the semi-circle qMQ, for which a heavy line is used.

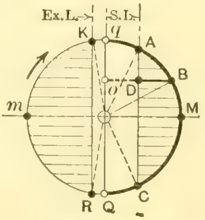

Fig. 397.

(**c**) Starting with the eccentric center at q in Fig. 397, the valve is evidently central; also it is moving to the right since the rotation is clockwise. When the eccentric reaches A, the valve displacement equals the steam lap, and *admission* begins; when at B, the valve is displaced a distance oB to the right and its steam edge is open an amount equal to DB; at M the valve has maximum displacement and maximum steam opening; at C the valve, now moving to the left, has displacement equal to the steam lap, hence *cut-off* is occurring; at Q the valve is central and its edges are overlapping by amounts equal to the respective laps; at R the valve is displaced to the left an amount equal to the exhaust lap, and is opening to *release*; at m the valve displacement is maximum to the left, and maximum exhaust opening is reached; and at K the displacement is equal to the exhaust lap, thus exhaust closure occurs and *compression* begins. The amounts of opening to steam and exhaust are shown by the lengths of the horizontal section lines. The diagram for the crank end of a symmetrical valve is exactly the same, except that it would be rotated through 180° (the minor events, Q, q, M, m, of course not being changed).

Fig. 397 may be called a **rectilinear diagram of valve displacements.**

(d) Note that admission and cut-off are controlled by the same valve edge (steam edge) but with valve motions opposite. This is apparent not only from Fig. 396, but can be seen from line AC in Fig. 397. Similarly, compression and release are controlled by the same edge (exhaust edge). Valve events controlled by the same edge may be called **conjugate events,** and it is important to note that changing the lap affects in opposite manner the two conjugate events which the edge controls. By means of a mental picture of the rectilinear diagram, and remembering the direction of rotation, one can readily visualize how a proposed change in lap will affect the events controlled by that edge, i.e., will show whether an event will be made to occur earlier or later.

POLAR DIAGRAM OF VALVE DISPLACEMENTS

(e) Fig. 398 shows a **polar diagram of valve displacements,** corresponding to the rectilinear diagram of Fig. 397. Given any eccentric position OE, the valve displacement $x = OB'$ is laid off as OB along OE. The locus of point B is $OAMCORmK$, which is composed of two circles. For displacements to the right the locus is shown by the heavy line, and that for displacements to the left by the light line. Given any eccentric position such as OE, the intercept OB is the valve displacement (here to the right).

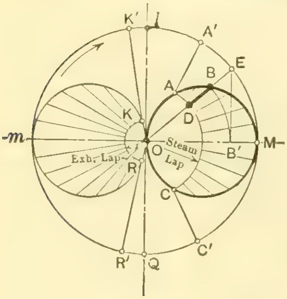

Fig. 398.

Arc AC is the *steam-lap line,* and is struck with the steam lap as radius. Arc RK is the *exhaust-lap line,* with radius equal to the exhaust lap.

When the eccentric arm is at Oq, the valve is central; when it extends through OA, the valve displacement equals the steam lap, and admission occurs; when at OM, the displacement and steam opening are maximum; when it passes through OC, cut-off occurs; at OQ, the valve is central; when through OR, release takes place; at Om, the displacement and exhaust opening are maximum; and at OK, compression begins. The lengths of the radial section lines show the amounts of opening of the steam and exhaust edges. The diagram for the crank end of a symmetrical valve would be exactly similar except that it would be rotated through 180°, the minor events, however, remaining unchanged.

310. Relative Valve and Piston Positions.—(a) While the foregoing diagrams are useful in showing the relative position of the valve with respect to its seat or to the *eccentric*, it is usually more desirable to know its position relative to the crank or piston positions. The relation between the eccentric, crank, and piston will now be discussed, considering the piston and valve to be both driven by Scotch yokes.

Referring to Fig. 399, when the crank is on head-end dead center, as shown by OP, the valve should have opened a slight amount, called the **lead** (principally in order to furnish steam to fill the clearance space and replace the vapor lost by initial condensation before the stroke starts), hence it will have a displacement equal to *lap plus lead*; and, further, it must be moving in the direction that will increase its opening. Obviously, when the crank is at OP the eccentric must be at OB for clockwise rotation, and at OB' when the rotation is opposite. In either case the angle $\alpha = qOB = QOB'$ is called the **angle of advance.** Therefore, in general, for a **direct-driven external valve** to have the lead and the proper direction of motion, the *eccentric must precede the crank by an angle equal to* 90 *degrees plus angle of advance.*[2]

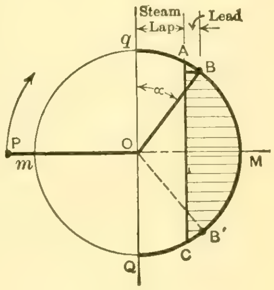

FIG. 399.

(b) Fig. 400 shows the successive critical crank positions during one complete revolution. These crank positions are located $90° + \alpha$ behind the corresponding eccentric positions that have been determined by means of the valve diagram. The figure also shows the development of the indicator diagram during the revolution.

311. Elliptical Diagram.—To show at a glance the simultaneous displacements of the valve and piston throughout the complete revolution of the engine, the displacements of the valve may be plotted as ordinates on the corresponding positions of the piston as abscissas. These coordinates can be obtained directly from the corresponding crank-pin and

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[2] The proper eccentric settings for internal valves and for indirect drives will be discussed in a later section.

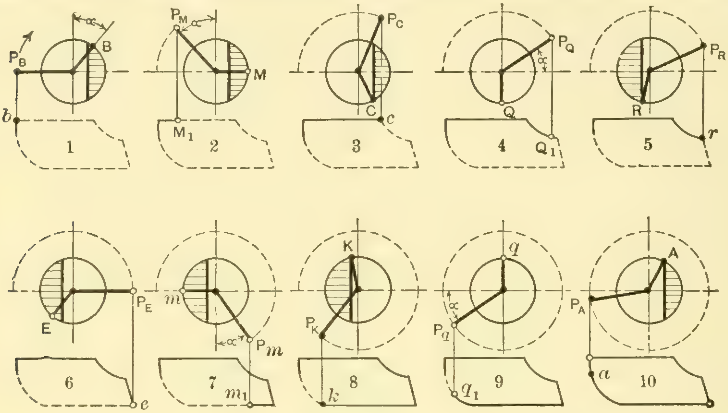

Fig. 400.

eccentric positions, as shown in Fig. 401 (*a*), or may be obtained from other valve diagrams which will be discussed later. With pure harmonic motion the resulting figure is an ellipse. In Fig. 401 (*b*), the valve displacements to the right are positive ordinates and those to the left negative.

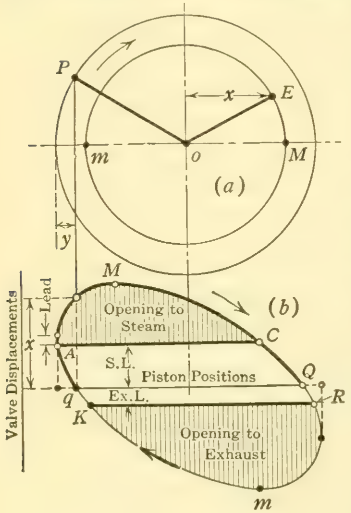

Fig. 401.—Elliptical Diagram.

Lines AC and RK are the head-end steam and exhaust lines and are drawn at distances from qQ equal to these respective laps. (If the exhaust lap is negative, RK will be above qQ.) The valve events are lettered in accordance with the notation adopted in Sect. **308** (**k**). The valve openings are shown by the lengths of the section lines. The opening at the beginning of the stroke is the lead.

If an indicator diagram for the head end were drawn just below Fig. 401, the piston positions for the valve events could be found by vertical projection.

As both ends of the valve have the same displacement, the same ellipse would be used for the crank end, but the steam lap would be located below qQ, and the exhaust lap above it (if positive).

The elliptical diagram shows at a glance the complete action of the valve, and shows how the valve opening varies with the piston positions. The part of the diagram above the steam-lap line may be considered as a **Diagram of Steam Openings.** Similarly, that part lying below *RK* is a **Diagram of Exhaust Openings.**

The piston and valve displacements used in constructing the elliptical diagram are more readily obtained from other valve diagrams, which will now be considered. The first two of these new diagrams are derived, respectively, from the rectilinear and polar diagrams previously considered, but show valve displacements corresponding to positions of the *crank* rather than relative to those of the eccentric, hence are more useful for many purposes than are the former diagrams.[3]

312. The Sweet Diagram.—In Fig. 402 (*a*) is given a rectilinear diagram of valve and eccentric displacements, similar to Fig. 397, and Fig. 402 (*b*) shows this diagram rotated backward (counter-clockwise) through an angle of 90° + α, which is the angle at which the crank follows the eccentric. The latter figure is known in this country as the Sweet diagram.[4] When the crank is at any location *OP* in Fig. 402 (*a*) and eccentric at corresponding position *OB*, the valve displacement is *oB* and its opening is *DB*. In Fig. 402 (*b*), with crank in same position *OP*, the distance *oB*, measured perpendicularly to *qQ*, gives the valve displacement, and *DB* is the valve opening. Thus the valve dis-

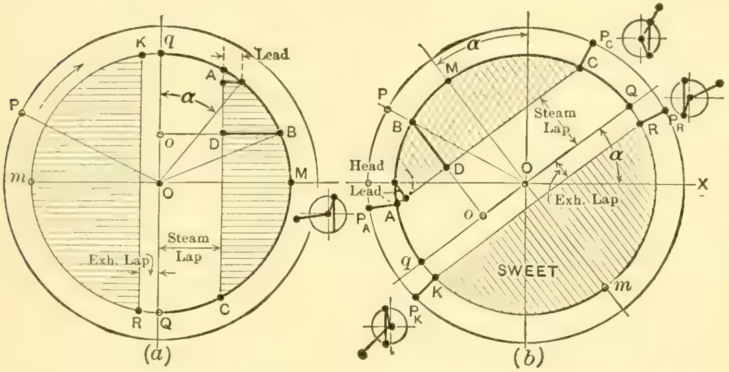

FIG. 402.—Sweet Diagram.

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[3] By using *crank angles* as abscissas and valve displacements as ordinates a *Sinusoidal Diagram* may be obtained. On it lap lines may be drawn, thereby obtaining intersections which determine the crank angles for the various events. However, this kind of diagram is not generally as useful as some others and hence its consideration will be omitted.

[4] It is called the Reuleux diagram in Europe.

placement and opening for any crank position can be obtained directly from the Sweet diagram without using eccentric positions.

In constructing the Sweet diagram for any valve and its eccentric, a circle is drawn with radius equal to the eccentric throw; the axis qQ is placed at angle α with OX; and lap lines AC and RK are drawn at distances from qQ equal to these laps. Then, OP_A is the crank position for admission; OM is that for maximum displacement; OP_C for cut-off; OQ, for valve central; OP_R, for release; OP_K, for compression. The openings are shown by the lengths of the section lines, and when the crank is on head dead center the opening is the lead. The foregoing is for the head end of a valve having positive exhaust lap. If the exhaust lap is negative, RK would be above qQ. For the crank end of the valve the steam lap would be located below qQ, and the exhaust lap, if positive, above that line. The little "Pilot Diagrams" show the relation of crank and eccentric for all valve events. The Sweet diagram is a very convenient one and hence is frequently used for analyzing the action of valves whose proportions and eccentric settings are known.

The elliptical diagram can be obtained from Fig. 402 (b) by using distances oB as ordinates which are erected on the vertical projections of the crank pin P.

313. Zeuner Diagram. In Fig. 403 (a) is given a polar diagram of

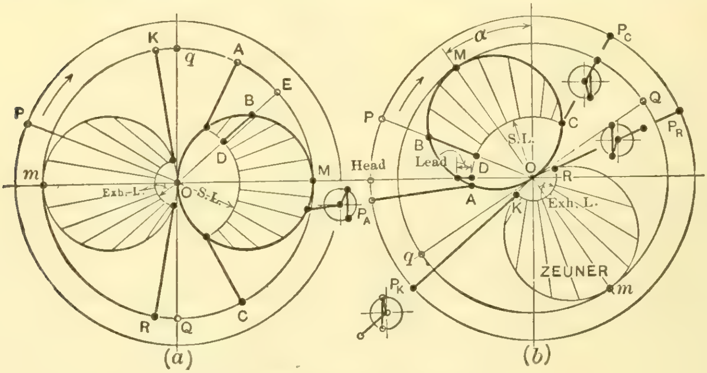

Fig. 403.—Zeuner Diagram.

valve and eccentric displacements similar to Fig. 398, and Fig. 403 (b) shows the same diagram rotated backward through the angle $90° + \alpha$. The latter figure is called a Zeuner diagram. In Fig. 403 (a), when the crank is at OP and eccentric at OE, the valve displacement is OB and its opening is DB. In the Zeuner diagram, with crank in the same position OP, OB is the valve displacement and DB is its opening. The crank positions for all events (major and minor) are shown and lettered on the figure, and the lengths of the radial section lines give the valve openings for the different crank positions. The lead is shown by the opening when the crank is on dead center. In Fig. 403 (*b*) the "Pilot Diagrams" show the relative positions of crank and eccentric at all valve events. Like the Sweet diagram, this one is widely used in investigating the functioning of valves when their proportions and eccentric settings are known.

The elliptical diagram can be easily obtained from the Zeuner diagram.

314. Bilgram Diagram.—**(a)** The foregoing diagrams are useful for *analyzing* the action of a valve when its dimensions and those of the eccentric are known. They are difficult to use, however, in *designing* a new valve gear, that is, in determining the valve laps and the eccentric throw and angle of advance which will give a proposed steam distribution in the cylinder, with specified widths of valve openings. The Bilgram diagram, which will now be considered, has the advantage that it can be readily used *either* for analysis or for design.

(b) Fig. 404 shows the principle of the Bilgram diagram. On it the line OQ is made equal to the eccentric throw and is at angle α (the angle of advance) with OX. Q is a *fixed* point on this diagram and is called the **lap-circle center.** The **Fundamental Principle** on which the construction and use of the Bilgram diagram are based may be stated thus: *The length of the perpendicular* (QD *in Fig.* 404) *from the lap-circle center* (Q) *to the crank* (OP), *produced if necessary, is the valve displacement corresponding to that crank position.* *Proof.*—When the crank is on dead center (at OP' in Fig. 404) the eccentric is at OE, and angle $EOY = \alpha$. Then if the crank rotates through angle β to OP, the eccentric moves through the same angle to B, and the valve then has displacement equal to $D'B$. Now angle $QOX = \alpha$ and $XOD = \beta$. Thus, if QD is drawn perpendicular to OP (produced), it is evident that triangles OQD and OBD' are equal and that $QD = D'B$; hence the perpendicular QD gives the valve displacement when crank is at OP, as stated in the "fundamental principle."

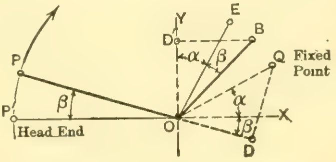

FIG. 404.—Bilgram Diagram.

The term "perpendicular" used in connection with the Bilgram diagram will hereafter be understood to refer to the length of perpendicular dropped from Q to the crank, produced if necessary.

The elliptical diagram can of course be constructed by using these perpendiculars as ordinates on piston positions as abscissas.

(c) Evidently the feet of the perpendiculars will be on a circle with *OQ* as diameter, as in Fig. 405. By subtracting the lap from the displacement perpendiculars, the openings of the valve are obtained.

In Fig. 405, with *Q* as center and radius equal to the steam lap, the **steam-lap circle** *BF* is drawn; hence the lengths of the section lines (drawn radially from *Q*) in this figure give the steam openings for the head end of the valve. Then *OA* (whose extension is tangent to the lap circle) is the crank position for admission, as the valve displacement (as shown by the length of perpendicular) just equals the steam lap. With crank at *OP* the opening is *FD* and displacement is *QD*; when the crank coincides with *OM* the opening is maximum and equal to *OB*; when at *OC* the opening is zero and cut-off occurs. When the crank is on head-end dead center the opening *L* is the lead.

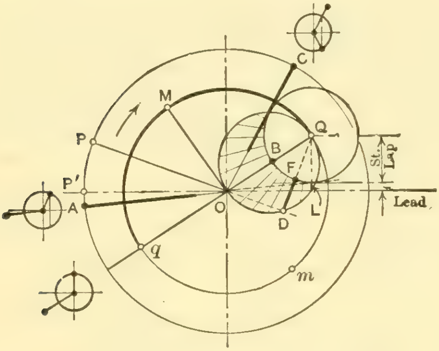

FIG. 405.

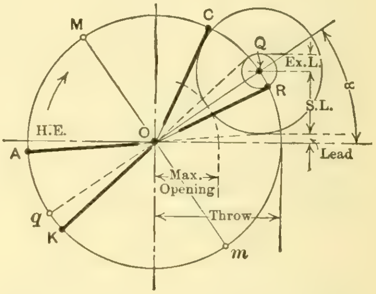

FIG. 406.

(d) Fig. 406 shows the completed Bilgram diagram for the head end of the valve, the eccentric circle being used also to represent the crank circle, as the crank angles are independent of the diameter of circle used. Compared with Fig. 405, it is seen that the smaller circle about *Q*, and the crank positions for the exhaust events, have been added. The small circle is the **exhaust-lap circle,** with radius equal to the exhaust lap. If *release* is to occur when the crank is in position *OR*, the exhaust-lap circle must be tangent to this line, for then the valve displacement (as shown by the length of the perpendicular) is equal to the exhaust lap.

When the crank coincided with *OQ* the valve was central, and, since in this case the exhaust opening does not occur until the crank has rotated clockwise past *OQ*, the valve is closed when central, therefore the exhaust lap is positive.

(e) The portions of the perpendiculars beyond the exhaust-lap circle represent exhaust openings. Evidently exhaust closure, or *compression*, takes place when the extension of the crank is tangent to the upper side of the lap circle; thus *OK* is the crank position for compression. When the crank coincides with *Oq* the valve is central; and, since in this case the exhaust lap is positive, the exhaust closure must take place before the valve reaches central position; hence *OK* is below *Oq* in this case, as the rotation is clockwise.

If the exhaust edge has **negative lap,** the crank position *OR* would be tangent to the upper side of the exhaust-lap circle, and the extension of *OK* would be tangent to the under side. When the lap is negative a broken line may be used for the exhaust-lap circle.

(f) The **application to design** problems requiring certain definite cut-offs, lead-openings, and maximum valve openings to steam, is carried out as follows: In Fig. 407, for the H.E. of the valve, starting with the X and Y axes, show the desired crank position *OC* for cut-off; draw a line (*L*) parallel to *OX* and above it at a distance equal to the specified lead; and with *O* as center and radius equal to the desired maximum valve opening strike an arc *B* in the position shown. From what has gone before, it is evident that the steam-lap circle must be tangent to these three lines. The location of its center *Q* can usually be found as quickly and as accurately by trial as by geometrical construction. Having the point *Q* determined and the steam-lap circle drawn, the diagram then shows the steam lap of the valve, and the throw and angle of advance of the eccentric, which must be used to obtain the desired results.

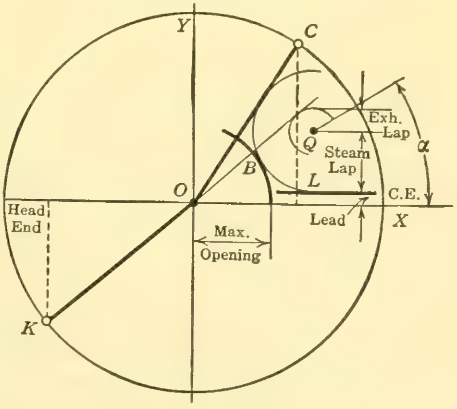

FIG. 407.

If *OK* in Fig. 407 is the desired crank position for compression, the exhaust-lap circle would be drawn tangent to the extension of this line, with center at *Q* just found, and its radius will equal the exhaust lap. Whether the exhaust lap is positive or negative can be determined in accordance with (e) in the foregoing discussion.

(g) For the **crank end** of the valve, the Bilgram diagram would be similarly constructed with lap-circle center at q, diametrically opposite Q. Fig. 408 shows the crank-end diagram drawn separately, with negative exhaust lap (shown dotted); it is usually, however, superposed on the diagram of the head end.

315. Distortion Due to Angularity of the Connecting Rod.—In Fig. 409, o is the middle of the stroke of the crosshead and the distance oO is equal to the length of the connecting rod aP. If an infinite rod is used, the displacement oa of the piston will of course be equal to the displacement OA of the pin P. If, however, a finite rod is used these displacements will not be equal. For, if the end a of the rod is kept stationary and the end P is uncoupled and swung to A', then oa will be equal to OA', which is seen to be greater than OA. It will be found that no matter where the crank is, A' will always be to the right of A. Since OA' is equal to the piston displacement, it is evident that, owing to the "**angularity**" of the connecting rod (of finite length), *the piston is always nearer the crank end of the stroke* than it would be with an infinite rod or Scotch yoke, except of course when the crank is on dead center. It also follows that: *The valve events occur later with respect to piston positions during the forward stroke and earlier in the return stroke* than they would with the Scotch yoke, but if the laps are equal, their mean is the same as the latter gives.

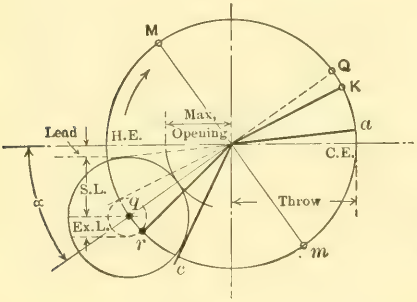

FIG. 408.

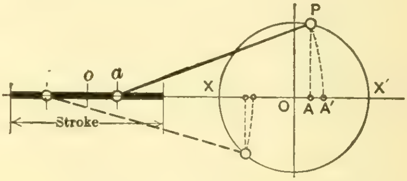

FIG. 409.

The distance AA' is the "distortion due to the angularity of the rod" and is equal to the difference between the length of the rod and its horizontal projection. This distortion is greatest when the crank is at right angles to the center line of the engine, and decreases to zero at the ends of the stroke. The shorter the rod when compared to the crank radius, the greater is this relative distortion.

If the diameter of the crank circle XX' is taken as representing the stroke of the piston, then, having any position such as A', the corresponding position of the crank pin P may of course be found by drawing the "connecting-rod arc" $A'P$; or if P is known at the start, A' may be found from it in a similar manner.

The **angularity of the eccentric rod** can be neglected in most cases, as this rod is usually very long when compared with the eccentric throw.

316. Valve Diagrams Considering "Angularity" of the Connecting Rod.—(**a**) The Sweet, Zeuner, and Bilgram valve diagrams all show the true positions of the *crank*; therefore, if the positions of the piston are not being considered, but only those of the crank, the angularity of the connecting rod would not be involved. If, however, after the crank positions have been found, the true positions of the *piston* are desired, it will then be necessary to include the effect of the angularity. Having already determined the crank positions, the corresponding true position of the piston would be found by drawing the connecting-rod arcs in the manner shown in Fig. 410 (*a*), (*b*), (*c*) for the Sweet, Zeuner, and Bilgram diagrams, respectively. Should the piston positions be known at the outset, then by drawing similar arcs the true crank positions can be found, and these would be used in constructing the rest of the diagram.

In the elliptical diagram, Fig. 410 (*d*), it is evident that the angularity causes all points on the ellipse, except those at the ends, to be displaced toward the crank end of the stroke. The resulting figure is of oval shape, in consequence of which the diagram is sometimes called the **Oval diagram.**

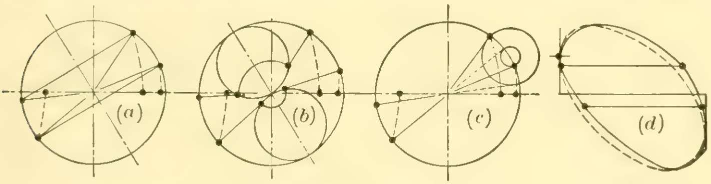

FIG. 410.

(**b**) Owing to the effect of the angularity of the connecting rod, the piston displacements for similar events in the two strokes of a double-acting engine will not be equal if the laps on the two ends of the valves are the same. It is possible to "equalize" the cut-offs by using unequal steam laps, but in that case the conjugate events (the admissions) are unequal. Similarly, the compressions can be "equalized" by using unequal exhaust laps, but then the releases will be unequal.[5] Equalization may also be accomplished by using special arrangements of rockers between eccentric rod and valve rod. This matter is discussed fully in most books that are especially devoted to valve gears, and will not be considered further here.

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[5] Except in the special case in which the exhaust lap would be zero if the "angularity" of the connecting rod were neglected.

317. Valve and Port Openings.—For the rate at which the steam is supplied to the cylinder at any instant to be equal to the rate at which volume is made available by the piston, the following expression must be satisfied:

$$av = AV, \qquad (398)$$

in which a = area of passage (sq. in., usually);
A = area of piston (same unit);
v = velocity of steam (ft./min., usually);
V = velocity of piston (same unit).

Then

$$a = AV \div v. \qquad (399)$$

As the piston velocity is variable, v also fluctuates.

Eq. (399) may also be used to find the **maximum area of valve opening** for a given engine. In that case a is the area desired, V is the *mean* piston speed (=2 × stroke × r.p.m.), and v is a constant having a numerical value which has been found satisfactory in practice. The value of v is usually between 6000 and 10,000 ft. per minute, depending upon the directness of valve passages and the class of engine. For simple H.S. engines v is usually taken as 8000 ft. per minute.

If the gear has a single valve, it is generally only necessary to see that the *steam* opening of the valve is sufficiently large, for the exhaust opening will always be more than is required because the exhaust lap is very much smaller than the steam lap.

The **width of the valve opening,** used in constructing the valve diagrams, is of course obtained by dividing the area of opening by its length. This length is nearly always equal to the length of port across the cylinder.

In case a simple valve is used, the **area of the port** in the cylinder should be sufficient for properly discharging the exhaust steam, as the same passage is used for both the entering and the outgoing vapor. Its area may be determined from Eq. (399), using for the coefficient of steam velocity (v) a value of from 4500 to 7000 ft. per minute, but about 6000 is usual in simple engines. This area is then more than is needed for the admission of steam, i.e., the steam edge of the valve does not usually open to the full width of the port.

If the exhaust passage is separate from the steam passage, this latter can have area about equal to the maximum valve opening to steam.

318. Cushioning the Reciprocating Parts.—(**a**) First, suppose there is no compression. Then when the piston approaches the end of its stroke the effective steam pressure and the inertia of the reciprocating parts are both acting towards that end of the stroke, taking up the slack in the bearings of the crankpin, crosshead pin and shaft in this direction. Now, when the steam is admitted on the other side of the piston, the pressure on the bearings is reversed. With reciprocating parts of small weight, and with high steam pressure, this reversal will be very sudden, and if there is much "play" in the bearings (and there must always be å little) the consequent impact or "hammering" will cause excessive stresses in the impinging parts, and will render the operation of the engine noisy.

(**b**) One method of preventing the occurrence of these undesirable features is to make the weight of the reciprocating parts so great that their inertia will oppose the pressure of the entering steam sufficiently to cause the play in the bearings to be taken up gradually, thus preventing impact. But, as the inertia forces are free ones which tend to move the engine on its foundation and to set up vibrations, it is usually desirable to have them small, even when counter-balancing is attempted; so this method is usually unsatisfactory.

(**c**) Another method is to arrange the valve to open gradually, say by having oblique edges on the valve, but this is accompanied by a more gradual cut-off, which is undesirable.

(**d**) Usually, the best method is to gradually reverse the pressure on the bearings by introducing compression; then, when admission takes place, there is no play to be taken up and consequently no impact.

It is possible to compute the inertia force of the reciprocating parts at the end of the stroke. Then in order to reverse the pressure on the bearings the steam pressure at the end of compression should equal or be greater than this inertia force plus the unbalanced steam pressure on the other side of the piston.

319. Early Valve Opening.—(**a**) If steam is admitted just as the new stroke begins, the pressure will not rise immediately to the value in the steam pipe because (1) the valve opens gradually, (2) the clearance space must be filled, and (3) a large proportion of the entering steam is liquefied by initial condensation on the cylinder. Hence it is necessary to have the valve open before the commencement of the stroke; that is, the valve is given "**lead.**" As the clearance volume is constant, the lead and crank angle at which opening occurs should be constant regardless of variations in cut-off, if the speed of the engine is uniform. The higher the speed and the less the compression, the earlier should the opening of the valve occur.

(**b**) In order to have the steam pressure drop to that of exhaust by the time the end of the working stroke is reached, the exhaust edge of the valve is also given lead, causing "**early release.**" As release and compression are conjugate events, the fixing of one of these events determines the other. Often it is not possible to have both occur as desired, in which case a compromise must be made.

320. Limitations of the Simple Valve.—It is impracticable to have cut-off occur early in the stroke with the simple D-valve because, in order to obtain a satisfactory width of opening in such cases, it is found (1) that the valve and eccentric are excessively large (and consequently the valve gear must work against great friction and inertia forces), and (2) that the release and compression occur too early in the stroke.

Fig. 411 is a Bilgram diagram for a valve cutting off at one-fourth stroke. If the scale is such that the maximum opening to steam represents one inch, it is seen that all of the foregoing statements are true. The simple D-valve is not used with cut-offs much earlier than five-eighths stroke.

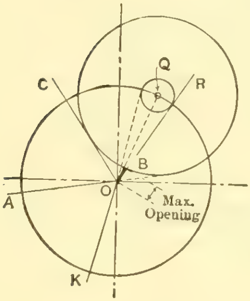


Fig. 411.

Ordinarily, the best economy in a simple engine is obtained when cut-off is at about one-fourth stroke; hence the simple D-valve should not be used when economy is important.

321. Special Types of Single Valves.—(**a**) By increasing the length of the steam edge of the valve a reduction can be made in the port width, laps, travel, and eccentric throw; but there are practical limitations to increasing the length of this edge in the simple flat valve.

(**b**) **Piston Valves,** Fig. 412, which may be considered as flat valves rolled into cylindrical shape, may have greater length of edge (equal to the circumference) than the simple flat valve, without having prohibitive size. Fig. 412 (*a*) shows an "external" piston valve; Fig. 412 (*b*) shows an "internal" one.

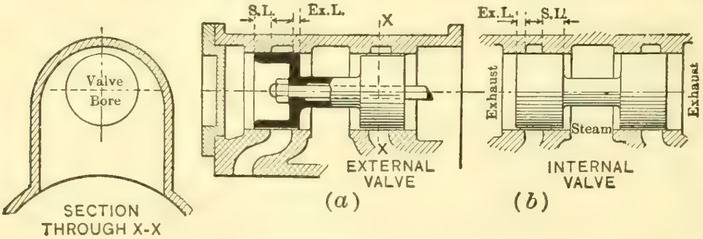


Fig. 412.—External and Internal Piston Valves.

(c) **Multiported Valves,** in which there are two or more working edges, are frequently used. Fig. 413 shows a **Double-ported Marine Valve,** each end of which has two steam edges and two exhaust edges. These valves are very heavy, hence on vertical marine engines their weight, and in some cases their inertia, is compensated for by means of a steam piston attached to an extension of the valve stem. The valve shown has a balance ring on its back which contains a vent that opens into the exhaust cavity.

(d) Some valves have **auxiliary ports** in them so arranged as to give multiported action. An example of this is the **Allen or Trick Valve,** shown in Fig. 414. This valve has an auxiliary passage *aa′* and valve seat so arranged that, as the valve moves to the right in the figure, the edge *f* opens simultaneously with the main steam edge *y*. The exhaust is single-ported.

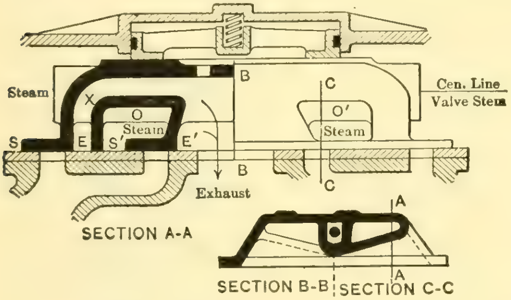


FIG. 413.—Double-ported Marine Valve.

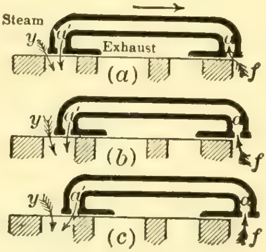


FIG. 414.—Allen Valve.

Considering the valve as moving to the right, the **phases of opening** of the steam edges are: (1) "*Double-ported*" action while edges *f* and *y* open at the same rate. This continues until the auxiliary port *a* is wide open. (2) With movement continuing, Fig. 414 (*b*), the opening at *y* increases but that through *a* remains constant, i.e., the opening is "*single-ported plus a constant.*" This phase of opening continues until *a′* in Fig. 414 (*c*) becomes throttled by the exhaust edge of the valve seat. (3) As the opening at *a′* is then decreasing (as the valve continues to the right) at the same rate as that at edge *y* is increasing, the effective area remains "*constant.*" (4) If the movement is sufficient to completely close *a′*, the valve becomes "*single-ported.*" Now if the valve returns to the left to close, the effective openings will decrease in the reverse order.

The openings of the steam edge of an ordinary valve are shown by the sectioned part above the steam-lap line of the elliptical diagram in

Fig. 401. This is also shown (somewhat distorted and without sectioning) by the light line in Fig. 415. The heavy lines in this figure show the character of the openings when an Allen valve, like that just described, is used. The numbers refer to the different phases of opening or closure. Note that the smaller openings are affected more than the larger ones.

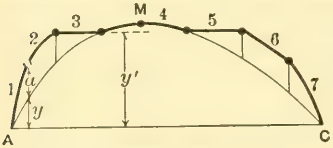
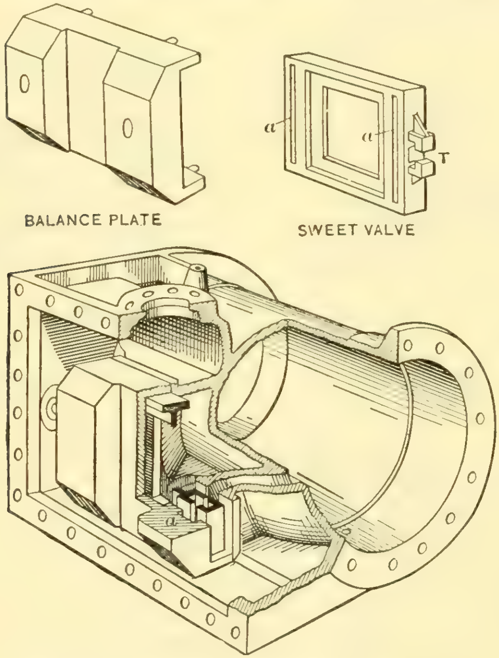


Fig. 415.

Piston valves similarly arranged have also been used (Armington-Sims valve).

(e) The **Sweet Valve** shown in Fig. 416 is another valve having an auxiliary port. It is a rectangular piston valve which slides between the valve seat and a **balance plate,** which part is supported by distance pieces so as to just clear the valve. The face of the balance plate has port edges, which correspond to those in the valve seat. All sliding surfaces are scraped to give sufficient clearance for free movement of valve but not enough to permit of appreciable leakage of steam. The valve has separate auxiliary ports *a* at each end, which causes double-ported action through at least part of the opening of the valve.


BALANCE PLATE

SWEET VALVE

Fig. 416.—Cylinder with Sweet Valve.

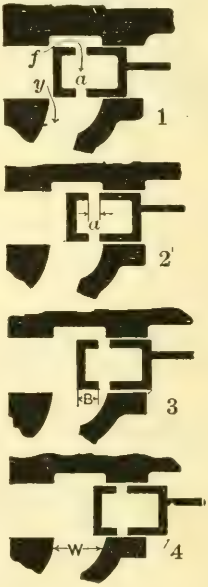


Fig. 417.

Referring to Fig. 417, it is seen that as the valve moves to the right the **phases of opening** of the steam edge are the same as those of the Allen valve; thus there is (1) "*double opening*," with y and f opening together; (2) "*single-ported plus a constant*," when f becomes greater than a; (3) "*constant*," when a is being closed by the exhaust edge of the valve seat at same rate that y is opening; and (4) "*single-ported*," when a is entirely closed by the exhaust edge. During closure the areas decrease in the reverse order.

The same auxiliary port, or another one, may be so arranged as to assist during the exhaust.

(**f**) A combination of the Allen and Sweet arrangements gives quadruple openings (Woodbury valve). There are many other forms of valves having auxiliary passages. For further discussion see textbooks specially devoted to Valve Gears.

(**g**) **Valve friction** is undesirable, as has been seen, not only because of the wear and waste of power it causes, but because it may disturb the action of the governor, if one is used. The whole back of the simple D-valve is subject to full steam pressure, while the larger part of the under side is exposed to the exhaust pressure. To reduce this unbalanced pressure, various schemes of **balancing** are employed, as already mentioned. The simplest is the use of a piston valve which is perfectly balanced except for its weight. The Sweet type of valve is practically the equivalent of the piston valve in this respect. Some valves have on their backs **balance** or **equilibrium rings** (like those shown in Figs. 413 and 418) so arranged that the area within the ring is subject to exhaust pressure and is about equal to the area having like pressure on the under side of the valve. However, in such cases there should always be enough unbalanced pressure to maintain steam tightness between valve and seat.

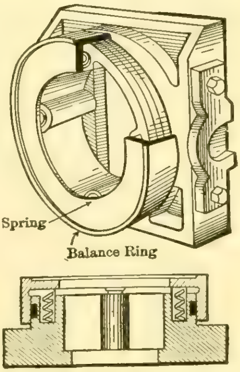


FIG. 418. — Valve with Balance Ring.

(**h**) In case there should be entrapped in the cylinder a quantiy of water more than sufficient to fill the clearance space, some part of the engine would break during compression unless some means of relief were provided. Sometimes **Relief Valves,** which are somewhat similar to boiler safety valves, are attached to the cylinder ends. Often the slide valve itself offers this relief; for example, the simple slide valve, with or without balance rings, and valves (like the Sweet) with pressure plates can lift from their seats and thus give relief. In such cases there should be springs or other devices to return the valve and balance ring, or plate, to its proper position after the cylinder has been relieved of the water. Also, the valve stem should not be rigidly attached to the valve for it would be bent when lifting occurs. Often the end of the stem is provided with a collar or T-head which fits in a pocket on the end of the valve with just sufficient looseness to permit the valve to lift. One of these pockets is shown at *T* in Fig. 416, and a similar one is illustrated in Fig. 418. Such an arrangement is further desirable as it permits the removal of the valve for inspection and its return without disturbing the adjustment of the valve rod. Piston valves offer no relief themselves, and engines with this type of valve should be provided with special relief valves or other similar devices on their cylinders.

322. Eccentric Settings and Guide Arrangements.—(a) It has been seen that if an *external valve* is *direct-driven* the eccentric is set at 90° + α ahead of the crank in the direction of rotation. The positions for *running over* and *under* in such a case are shown at (1) and (2), respectively, in Fig. 419. If the valve is *internal*, its displacement and motion must be just opposite those for the external one; hence the eccentric position is always diametrically opposite that for the latter case, and therefore is placed 90° − α behind the crank, as is shown at (3) and (4) in the chart, respectively, for running over and under.

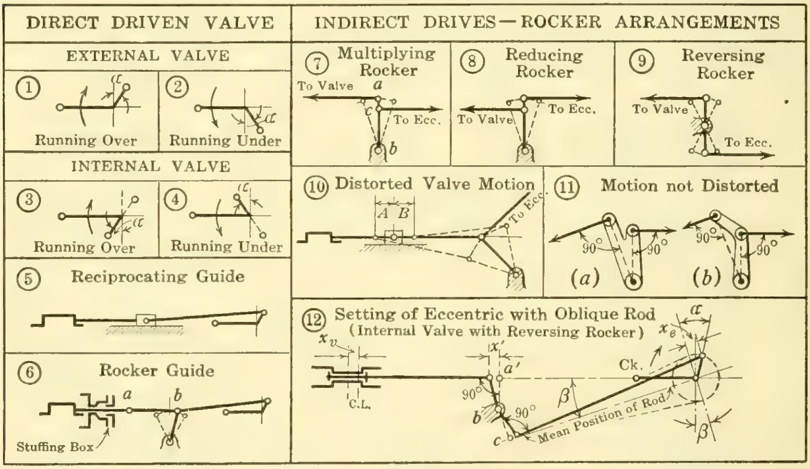


FIG. 419.—Eccentric Settings and Guide Arrangements.

(b) At (5) in the chart the eccentric rod end is shown connected to a pin on a *reciprocating guide*, or *slide a*, to which the end of the valve stem is attached. This slide is of course guided in the same manner as is a crosshead. At (6) the eccentric rod end *b* is guided by a *rocker*, and as *b* does not move in a horizontal line it is usually necessary to provide a flexible coupling *a* in the valve stem connection. This coupling may be a knuckle joint or a ball-and-socket joint, relying on the stuffing box to properly guide the valve stem, or it may have a guide similar to that in (5).

(c) With the arrangements shown in (5) and (6) the valve displacement is always equal to that of the eccentric, and the valve is said to be *direct-driven*. At (7) is shown a *multiplying rocker*, hence for a given valve travel the eccentric throw would be less than that with direct drive, in the ratio cb/ab. With a *reducing rocker*, such as that shown at (8), an eccentric throw larger than that for a direct drive would be used. At (9) is shown a *reversing rocker*, which may be arranged to give equal (but opposite) displacement, at the two ends, or it can give multiplied or reduced motions. With such a rocker the eccentric would be located diametrically opposite the position it would have with a direct drive.

(d) With the arrangement shown at (10) there is *distorted valve motion* because, with equal angular movements of the rocker on either side of its middle position, the valve displacement A will be less than displacement B. Obviously, **to prevent distorted motion** from occurring *the angles between the rods and the center lines of their respective rocker arms should be* 90° *when the valve is central.* Arrangements of rockers that fulfill this requirement are shown at (11).

(e) With an **oblique eccentric rod** and with a direct-driven external valve, or with an internal valve with a reverse rocker as in (12), the eccentric is not placed $90° + \alpha$ ahead of the crank, but is at that angle ahead of the *mean position of the eccentric rod when the crank is on dead center.* This is obviously the correct setting with this arrangement, for only then is $x_v = x_e =$ lap + lead, when the crank is at Ck. Thus, if the eccentric rod is inclined downward, as in (12), at an angle β with the center line of the engine, the angle at which the eccentric leads the crank is $90° + \alpha - \beta$.

(f) Although distorted valve motion is usually avoided, it may sometimes be desirable, as will be seen later, to introduce it to a small extent in order to "equalize" valve events. Also, with Corliss and certain other valve gears in which there are separate valves for each end of the cylinder, considerable distortion is used in order that the valve will have but little motion after closure (hence a minimum of friction and wear) and wide and rapid motion after opening. These cases will be considered later.

323. Valve Gears for High-Speed Single-Valve Engines.—(a) The H.S.S.V. engine was briefly described on page 151. It is fitted with a "shaft governor" which controls the point of cut-off by varying the position of the eccentric with respect to the crank.

(**b**) Simple engines of this type usually have cut-off occur at about one-fourth stroke when operating under "normal load." The *range of cut-off* is generally from zero to five-eighths or three-fourths stroke, and the *range of load* is from "friction load" to 50 or even 100 per cent overload.

It was shown on page 212 that if a simple D-valve is used to give cut-off as early as one-fourth stroke, certain features will be introduced which are undesirable in the ordinary case. Two of these are early release and early compression.

(**c**) It so happens, however, that the *early occurrence of the exhaust events* is desirable in a high-speed engine. The early release permits a more perfect toe to be formed on the indicator diagram in the shorter time available with high speeds; and the high compression is needed to overcome the greater inertia of the reciprocating parts so as to reverse the pressure on the bearings and pins before admission occurs.

An excessive pressure at the end of compression can be prevented by increasing the ratio between clearance volume and piston displacement. With a given clearance volume and piston area, this ratio can be made large by using a short stroke; and with a high rotative speed a short stroke is desirable in order to keep the piston speed within safe limits. Thus, what are faults with low speeds are necessities when the speeds are high, and are the things that make the short-stroke, high-speed engine possible.

(**d**) The *excessive size of the valve-gear parts*, which ordinarily occurs when a D-valve is designed for an early cut-off (as was shown in Sect. **320**) may be overcome by increasing the length of port. This permits using a narrower valve opening, reduces the laps and travel of the valve and the size of the eccentric, and decreases the friction and wear of the valve gear. The greater length of port may be obtained by using a wide valve, a piston valve, or a multiported valve.

With the type of valve gear used on this kind of engine, the travel of the valve varies with the cut-off; thus the earlier the cut-off the more restricted is the opening of the valve. The valves may be designed to have openings ample for the latest cut-offs, and to have auxiliary ports arranged in such a manner as to assist during the narrower openings only, and to have little or no effect on the wider openings. Examples of these various types of valves have already been given.

Sometimes *special arrangements of linkage* are employed to give wide openings with small travel, as in the case of the "High-Speed Corliss" engine.

(**e**) The friction of the valve is undesirable, not only because it decreases the mechanical efficiency of the engine and causes wear, but also because it disturbs the action of the shaft governor. The latter is especially true if the governor is of the Inertia type. The governor is also affected by the inertia of the valve gear. Hence high-speed engines use valves that are balanced and are of light weight.

(f) In Fig. 420, which is a diagram of positions, the line *1-2-3* is one path, with respect to crank OP on dead center, over which the eccentric might be shifted by a shaft governor in adjusting the cut-off to meet the power demanded of the engine; and a_1c_1 is the steam-lap line. When in Fig. 420 (*a*), the eccentric is at *1* (with throw *O–1* and angle of advance α_1) the cut-off is at about three-fourths stroke, for when the eccentric has rotated in the direction of the arrow to c_1 (displacement equal to steam lap) the crank pin is at C_1; when, in Fig. 420 (*b*), the eccentric is at *2* (with throw *O–2* and angle of advance α_2) the cut-off is at about one-half stroke, for when the eccentric is at c_2 the crank pin is at C_2; and when the eccentric is at *3*, diametrically opposite the crank, the cut-off is at C_3. In this case the path *1–3* is so selected that it will coincide with La_1 when the crank is in position OA_1; hence the crank-pin positions (A_1, A_2, and A_3) for the admissions corresponding to all cut-offs will coincide, that is, the admission is constant.

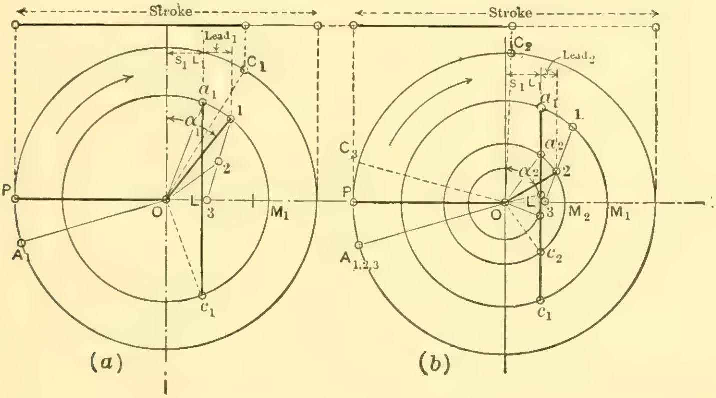


FIG. 420.—Diagram of Positions.

With crank at OP, the horizontal distance between the eccentric and the steam-lap line, a_1c_1, is the lead. From the figure it is seen that Lead_2 is less than Lead_1, and that as the eccentric is shifted to give earlier cut-off the lead becomes less.

Fig. 420 (*b*) also shows that when the eccentric throw is *O–1* the maximum valve opening is LM_1, that when the throw is *O–2* the maximum opening is LM_2, and with throw equal to *O–3* this opening is *L–3*. Thus, the maximum openings decrease as the cut-off is made to occur earlier. The valves are therefore usually designed to have proper opening at latest cut-off when operating as a single-ported valve and to be multiported when early cut-offs occur.

If the exhaust-lap line is added to the diagram of positions, such as Fig. 420, and the crank positions are determined for the exhaust events that correspond to the different points of cut-off, it will be found that as cut-off occurs earlier, the release and compression also occur earlier.

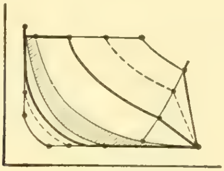


Fig. 421.

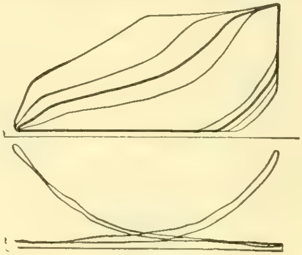


Fig. 422.

Thus with early cut-off there is greater compression than with late cut-off. Fig. 421 shows how these events vary and affect the form of the conventional indicator diagram; and Fig. 422 are actual diagrams obtained from a high-speed engine.

From the foregoing, the following general statement can be made: *As the eccentric is shifted from the outer to the inner position, cut-off, release, and compression are made to occur earlier and the maximum opening is decreased.*

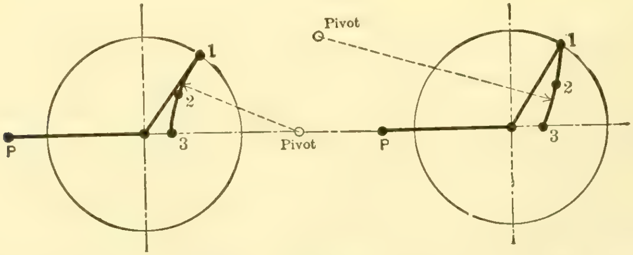


Fig. 423. Paths of Swinging Eccentric. Fig. 424.

Instead of shifting the eccentric over a straight path, it can be swung about a pivot in the governor case or flywheel. When a governor of the *inertia* type (Fig. 392) is used with an external valve, the eccentric center must be moved over a circular path with pivot opposite the crank, as in Fig. 423, in order to have the inertia of the governor weight-arm act in the proper direction. When the ordinary centrifugal governor is used, the eccentric may be pivoted on either side of the shaft with respect to crank, hence its path may be either that in Fig. 423 or that in Fig. 424. These curved paths are usually made to approximate the straight one of Fig. 420 as closely as the arrangement of the mechanism will permit. With curved paths the crank angle for admission will not be quite the same for all positions of the eccentric.

(**g**) The various valve-gear diagrams can be constructed in the usual manner, taking each position of the eccentric independently.

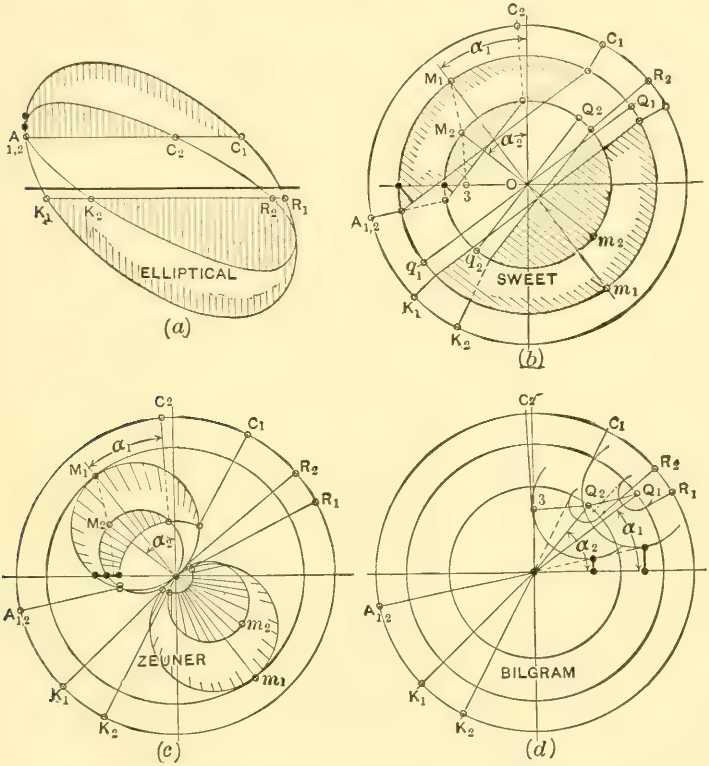


FIG. 425.—Valve Diagrams for Variable Eccentric Valve Gear.

The diagrams for the different eccentric positions may be drawn separately, but usually they are superposed on one another, as in Fig. 425, which corresponds to a path similar to that shown in Fig. 423.

(**h**) At any *one* cut-off the valve events can be equalized in the manner mentioned in Sect. **316** (**b**). In some cases it is possible to approximate the equalization in all positions of the eccentric by using a special arrangement of rocker arm or guide for the eccentric-rod pin.[6]

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[6] See Halsey's "Steam Valve Gears."

324. General Characteristics of Independent Cut-off Gears.—(**a**) If an engine operates at a constant speed, as most engines do, it is desirable to have the admission, compression, and release remain constant, no matter how the cut-off varies. The single valve with shifting eccentric does not give this desired constancy of all these events, nor does it give sufficient opening and sharp closure when cut-off is early in the stroke. If, however, instead of one valve, two or more are used, with *two independent sets of valve gear*, it is possible to avoid some or all of these difficulties. Owing to the complications and extra expense involved with such arrangements, they are not often used on the small high-speed types of engines. They have been quite commonly applied, however, on larger- and longer-stroke, medium-speed engines.

(**b**) In general the gear having the two valves may be arranged in either of two ways:

1. Each valve may control one pair of conjugate events at each end of the cylinder; thus, one valve, having a fixed eccentric, may operate the releases and compressions, while the other valve takes care of the admissions and cut-offs, the latter events being changed by shifting the eccentric by means of a shaft governor, as in the valve gear used on high-speed engines. In this case, separate valve diagrams would be drawn for each pair of conjugate events: that for the steam events would be constructed the same as for the simple shifting-eccentric gear, and that for the exhaust events would be drawn the same as for any case in which the crank positions for the opening and closing of the exhaust edge are given, together with the maximum width of opening desired. This arrangement will not be considered further.

2. In the other arrangement of the gear, one valve, which will be termed the *main valve*, controls the admission, compression, and release, and is driven by a fixed eccentric ("main eccentric"). The other valve operates the cut-off only, and will be called the *cut-off valve*. It is an *intercepting valve*, being located between the main valve and the source of steam supply. It may slide on a separate valve seat, or it may ride on the back of the main valve, in which case it is called a *"riding" cut-off valve.*

As will be seen later, the variation in cut-off may be accomplished by changing either the lap of the cut-off valve, or the position of the cut-off eccentric with respect to the crank. The range of cut-off on medium-speed engines having this valve gear is usually from zero to five-eighths or three-fourths stroke. Under normal load simple engines of this type usually cut-off at about one-fourth stroke.

(**c**) In *all* the gears having a cut-off valve of the intercepting type, the **main valve** or valves (operated by the main eccentric) control the admission, release, and compression. In order to determine the proportions of the main valve and its eccentric, the procedure is as follows:

Referring to Fig. 406 it is seen that the line OQ bisects the angle formed between OR and OK (produced); hence the two conjugate events R and K, which can be decided upon initially, fix the angle of advance of the main eccentric. Thus, in the *Bilgram diagram,* Fig. 426, OQ_1 is made to bisect the angle between R and K (produced) and this determines α. Then, after drawing the lead line L, and the arc W for maximum opening, the steam lap is determined by drawing the lap circle tangent to W and L in the figure, the center Q being on OQ_1. Having located Q, the exhaust-lap circles can then be drawn. The H.E. diagram is now complete and all dimensions for that end of the valve and the eccentric have been found. The cut-off of the main valve is unimportant, provided it is at least as great as the latest to be given by the cut-off valve. Compression, of course, can be equalized by using unequal exhaust laps.

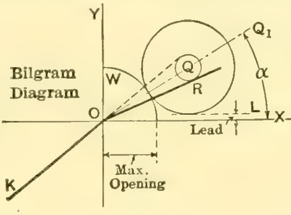

FIG. 426.

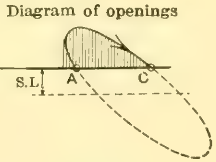

FIG. 427.

The *diagram of openings* of the main valve, which is that part of the elliptical diagram which lies above the steam lap, $S.L.$, is shown in Fig. 427 for the H.E. of the valve.

(**d**) In *all* valve gears in which there is a separate cut-off valve it is necessary that this valve open before the main valve does, as the latter controls the admission. To provide for this, it is often essential to have negative lap on the cut-off valve and large angle of advance, α', for the cut-off eccentric (α' may even be greater than 180° in some instances).

In the construction of the various valve diagrams for the **cut-off valve** (or any valve) with *negative lap* and *large angle of advance* the principles are identical with those previously described: In each case the angle of advance is located in exactly the same manner as before, and the negative lap is merely laid off opposite to positive lap; the openings are then equal to the displacement plus the negative lap, and the closures equal the displacement minus this lap. With negative lap the valve is open when it is central, hence closure must occur after the central position has been passed, and opening takes place before that position is reached. Figs. 428 (b) to (e) show the various valve diagrams with an angle of advance α' greater than 90°, and Fig. 428 (*a*) gives the actual position of the eccentric with respect to the crank when the latter is on dead center. The lengths of the section lines show the widths of valve openings.

The generating point in the elliptical diagram will move around the ellipse in a counter-clockwise direction if the angle of advance is greater than 90°, as it is in Fig. 428. As before, the part of the elliptical diagram lying above the steam-lap line constitutes a *diagram of openings* for that valve.

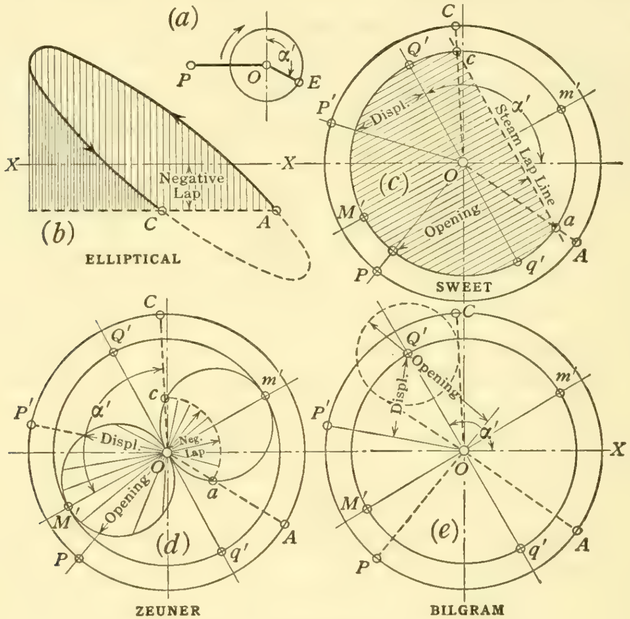

FIG. 428.

The cut-offs at one point in the stroke can be equalized by using unequal laps.

(**e**) Referring to Fig. 428 (*e*), the cut-off can be varied either (1) by altering the size of the lap circle (which may even be made positive), or (2) by changing the angle of advance, α'. In the latter case a shaft governor may be used to change automatically the angle of advance by turning the cut-off eccentric about the center of the shaft on which it is loosely mounted. Both methods are used in practice. A combination of them is possible, but not commercially feasible.

325. Independent Cut-off Valve with Stationary Seat.—Fig. 429 shows diagrammatically an arrangement with cut-off valve *C* riding on an *independent stationary valve seat* which is located between the main valve M and the steam pipe. The main valve, which is driven by a fixed eccentric, controls the admission, release, and compression, and is designed in the manner outlined in Sect. **324** (**c**). It has a false end B so that the space beyond will not be included in the clearance. The cut-off valve is driven by an independent eccentric and controls only the one event. The cut-off can be changed in either of two ways already mentioned.

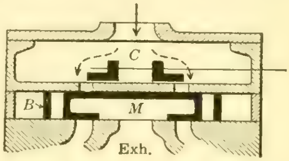

FIG. 429.

Case I. Variable C.O. Lap.—The lap of the cut-off valve may be altered by any of the following methods:

1. The valve may be in two parts, which are mounted on the valve stem with R.H. and L.H. threads, respectively as in Fig. 430. Then, by turning the stem, the distance between the ends, and consequently the laps, can be varied. With this arrangement the adjustment is made by hand, and the point of cut-off for the setting can be read on the indicator, which is moved by a non-rotating nut on the valve stem. (It is difficult to arrange a governor to make the adjustment in this case, as several revolutions of the valve stem are required to accomplish the full range of cut-off.)

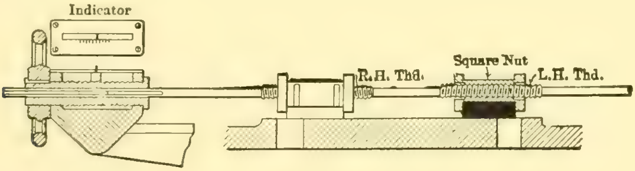

FIG. 430.

2. Fig. 431 shows the back of another arrangement in which the edges of the C.O. valve and ports are oblique. On the back of this valve is a rack with which a pinion on the valve stem engages. By turning the stem the valve may be raised or lowered (as viewed in the figure), thus changing the distance between its edge and that of the port.

3. Fig. 432 shows a somewhat similar arrangement, except that the valve face and seat are cylindrical surfaces. The valve is fastened to the stem, so that by turning the latter the lap is changed.

The arrangements shown in Figs. 431 and 432 can be controlled by a spindle governor, which can be connected to an arm on the valve stem. The valve diagrams for this case can be constructed by the methods given in the preceding section.

Case II. Variable Angle of Advance.—Fig. 433 shows how the angle of advance of the cut-off eccentric can be varied to change the cut-off, the lap of the cut-off valve remaining unchanged. In this figure M is the main eccentric; points 0, $\frac{1}{4}$, and $\frac{3}{4}$ on the circle are the positions of the cut-off eccentric for those cut-offs; α_0' is the maximum angle of advance of the cut-off eccentric, $\alpha'_{\frac{3}{4}}$, is that for three-fourths cut-off; and λ is the *range angle* through which the governor must turn the eccentric on the shaft.

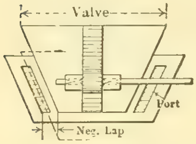

Fig. 431.

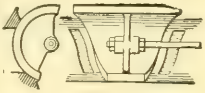

Fig. 432.

Fig. 434 shows the corresponding diagram of openings of the cut-off valve (dotted lines) superposed on that for the main valve (heavy lines), and the sectioning shows the effective opening from the time of admission of the main valve to the closure of the cut-off valve at one-fourth stroke.

The valve diagrams for this case are constructed in the manner outlined in Sect. **324** (**c**) and (**d**).

326. Riding Cut-off Valves.—(**a**) Instead of having a separate seat for the cut-off valve, this valve may ride directly on the back of the main valve (or within it, if a piston or box valve is used) and perform its functions with respect to a port in that valve. There are several such arrangements possible. One, the **Buckeye Gear,** is in effect the exact equivalent of the arrangement described as Case II above.

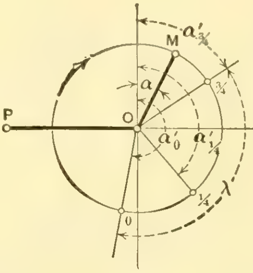

Fig. 433.

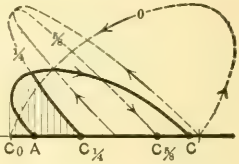

Fig. 434.

The arrangement of valves in this gear is shown in principle in Fig. 435. At (*a*) the main valve is shown as open to exhaust, and at (*b*) both valves are central. The main valve is a box filled with live steam, thus it is practically a reciprocating steam chest. It is an internal valve, taking steam from the inside and exhausting at the ends. The cut-off valve is an external valve, which rides inside of the main valve. It opens and closes with respect to a port edge in the main valve and has negative lap of constant amount relative to that edge. Its valve stem passes through that for the main valve, the latter being hollow.

The main valve is driven by a fixed eccentric, and controls admission, release, and compression. The cut-off valve is driven by an eccentric which is controlled by a shaft governor which turns the eccentric about the center of the shaft, thus varying the angle of advance, α', as in Fig. 433.

(**b**) The arrangement of the rockers which guide the eccentric-rod ends is shown diagrammatically in Fig. 436. The main rocker *ab* is pivoted at *b* to the frame of the engine. The cut-off rocker *cd* is pivoted at its middle *e* to the middle of the main rocker. With this special arrangement of rockers, the displacement (δ) of the center of the cut-off valve with respect to the center of the main valve is given by the distance between *a* and *c*, which is the same as that between *b* and *d*. Evidently, then, the motion of the cut-off valve with respect to the main valve is the same as that of *c* with respect to *a*, or opposite to that of *d* with respect to *b*. Since *b* is a *fixed point, the motion of the cut-off valve with respect to the main valve is the same as that of a simple valve with respect to a fixed seat.* The distance the cut-off valve travels with respect to the main valve remains constant, no matter how the cut-off and phase relations of the two valves are altered by changing the angle of advance of the cut-off eccentric. In Fig. 436 the cut-off eccentric rod is shown horizontal and the rocker *ced* is straight, whereas in the actual case this rod is oblique and the rocker is bent. The true arrangement of the cut-off linkage is similar to the mechanism shown at 12 in Fig. 419, except that the cut-off rocker is pivoted on the main rocker.

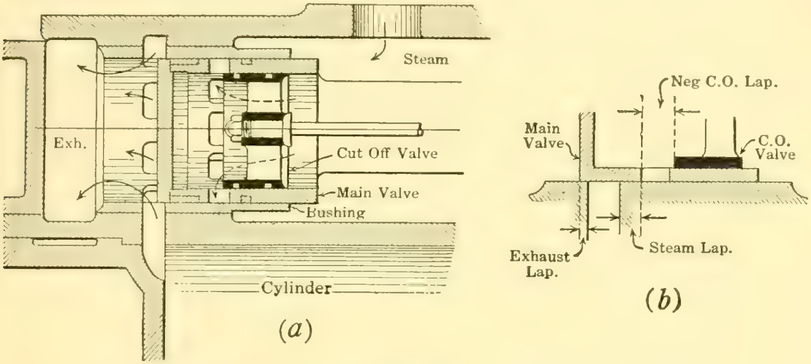

FIG. 435.

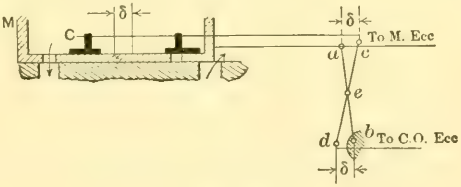

FIG. 436.

The setting of the cut-off eccentric is also that shown in the same figure.

Obviously this arrangement is equivalent to Case II in its results, and may be analyzed by constructing the valve diagrams in the same way as for that case.[7]

(c) In other "riding cut-off" gears, this peculiar arrangement of rockers of the Buckeye gear is not used, hence both valves receive motion direct from their respective eccentrics. The general arrangement of the valves for such cases is shown in Fig. 437, in which both valves are external. The main valve has a false end B, the purpose of which is to provide an edge F, with respect to which the cut-off valve opens or closes. Each valve is driven by its own eccentric, the location of which is shown in Fig. 438. The main valve controls the admission, release, and compression, and is designed as in (c) of Sect. **324.** To analyze the action of the cut-off valve, its motion *with respect to the moving main valve* must be considered.

(d) As the crank revolves, not only do the eccentrics revolve about the center of the shaft, but they rotate about each other. That this last statement is true can be seen by turning Fig. 438 about O, and it will be noticed that, at the same time, the cut-off eccentric rotates about the main eccentric and in the same direction as that in which the crank turns. Evidently, then, *the motion of the cut-off valve with respect to the main valve is produced by the rotation of the cut-off eccentric about the main eccentric;* hence, *this motion is equivalent to that which a simple valve would have, with respect to a fixed seat, if driven by an imaginary eccentric having throw equal to the distance between the eccentric centers* (Fig. 438) *and with angle of advance equal to* θ. This imaginary eccentric will be called the **relative eccentric,** R. The method of determining its location with respect to the crank and the actual eccentrics is shown in the figure. Then, to analyze the action of the cut-off valve with respect to the main valve (its moving seat), the position of the relative eccentric

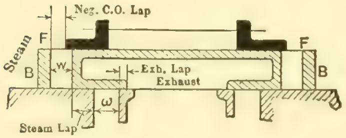

FIG. 437.

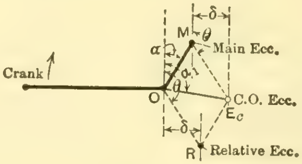

FIG. 438.

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[7] The complete analysis of the Buckeye gear affords an excellent exercise in constructing the valve diagrams (including the diagram of openings) and in laying out rockers and locating eccentrics.

must first be determined, after which the valve diagrams would be constructed in the usual manner, but using the throw and angle of advance of this eccentric.

(e) The **Meyer Valve Gear** has the same arrangement of valves as that shown in Fig. 437. The cut-off eccentric is fixed and usually has a 90° angle of advance, that is, it is located opposite the crank. The cut-off is varied by *changing the lap* of the valve by the method shown in Fig. 430.

(f) The **Russell Valve Gear** is similar in character to Figs. 437 and 438, but the cut-off is altered by *changing the angle of advance* of the cut-off eccentric. This adjustment is made by a shaft governor which turns the eccentric on the shaft.

The valves are shown in Figs. 439 and 440. The main valve operates the admission only. The exhaust is controlled by separate triple-ported valves of the Corliss type, as shown in Fig. 440. Shifting the cut-off eccentric changes both the throw and the angle θ of the relative eccentric. This throw is small when the cut-off eccentric is in position for the late cut-offs, hence the necessity of making the C.O. valve triple-ported.

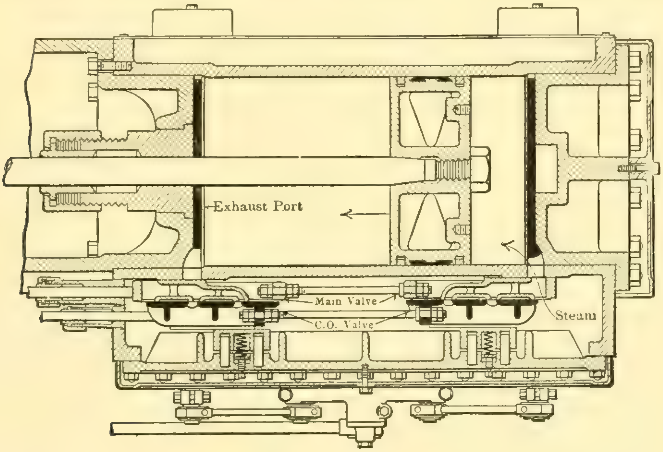


FIG. 439.

(g) The **McIntosh-Seymour Gear,** shown in Fig. 441, also has separate main, cut-off, and exhaust valves, but these are of the gridiron type which require but little amplitude of movement to obtain the desired openings. The six valves are driven by oscillating rockers and toggles in such manner that after the valves are closed they have little motion.

The main and exhaust valves are driven by a fixed eccentric, and the cut-off valve by an eccentric whose angle of advance is varied by a shaft governor, the arrangement being similar to that in Fig. 433. The rocker shafts *B* and *C* in Fig. 441 are given an oscillatory motion by the eccentrics acting through bell cranks.

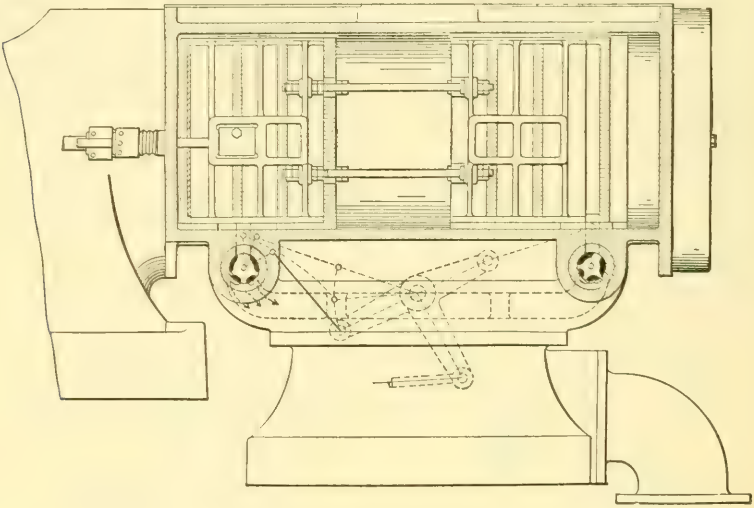


FIG. 440.

In Fig. 442 the distorted part of the elliptical diagram for closure of the main valve is shown dotted, while the opening diagram is shown by bold lines. Superposed on the latter are lines showing the closure of the cut-off valve. Fig. 443 gives the distorted elliptical diagram for the exhaust valve. The valve movements after closure are much less than with ordinary valve gears.

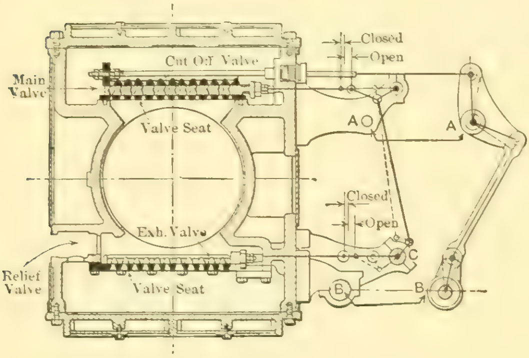


FIG. 441.

(h) There are many other possible arrangements of riding cut-off gears, a great number of which are in actual use.

327. Gears with Oscillating Valves.—(a) Instead of having the slide valve flat, it may have a curved face, as in Fig. 444, in which case the valve oscillates about center O'. The displacement x of the eccentric-rod pin U from the Y-axis is always equal to that of the eccentric with respect to the vertical axis through the shaft. This valve is substantially equivalent to the ordinary flat D-valve, and would be designed or analyzed in the same manner, using the same valve diagrams. These diagrams show the true positions of the crank for all events; but the laps, displacements, and openings are chordal, that is, would be measured as chords instead of as arcs.

(**b**) This arrangement of valve introduces very long steam passages, as they extend from the middle to the ends of the cylinder, and this is not conducive to economical performance, as has already been seen.

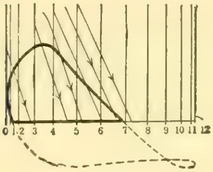


FIG. 442.

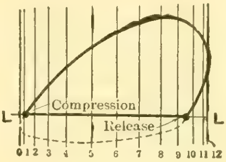


FIG. 443.

(**c**) A better arrangement in this respect is one in which there are *four* oscillating valves, as in Fig. 445, each of which performs the single function of one of the four edges of the single valve. In the figure the outer edges of the two upper valves control the steam events, and the corresponding edges of the lower valves operate the exhaust events. The other edges of the valves perform no function except to remain closed. The chordal laps would be the same as in the case of the single valve of Fig. 444. The valves shown in Fig. 445 are of the "Corliss" type, and, as already noted, with this arrangement the steam and exhaust passages are very short and direct, thus the clearance volume and surfaces are relatively small.

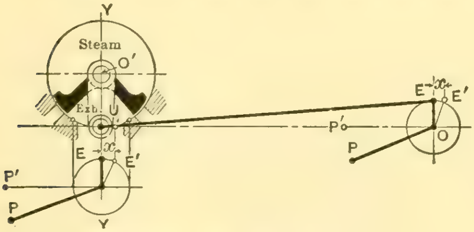


FIG. 444.—Oscillating Valve.

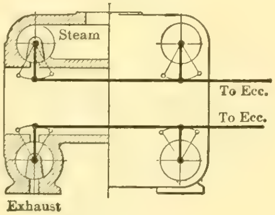


FIG. 445.

(**d**) All four valves may be driven by the single variable eccentric with shaft governor, as is common with high-speed engines. It is better, however, to connect the steam valves in this manner, and to drive the exhaust valves by a separate fixed eccentric, so that release and compression will remain constant.

(e) The arrangement just considered has, however, the same fault as others having direct-driven valves, in that the valves have large movement after they have closed. Not only is it advantageous to have this motion cease as soon as the overlapping of the valve is sufficient to prevent leakage, but it is also desirable to have more rapid movement of the valve after it opens than is obtained with the simple gear. Both results can be effected by using distorted motion, which may be produced by special arrangements of links and rockers. One such arrangement is shown in the upper right-hand corner of Fig. 446. *High-speed Corliss Valve Engines* use valve gears of this general type.

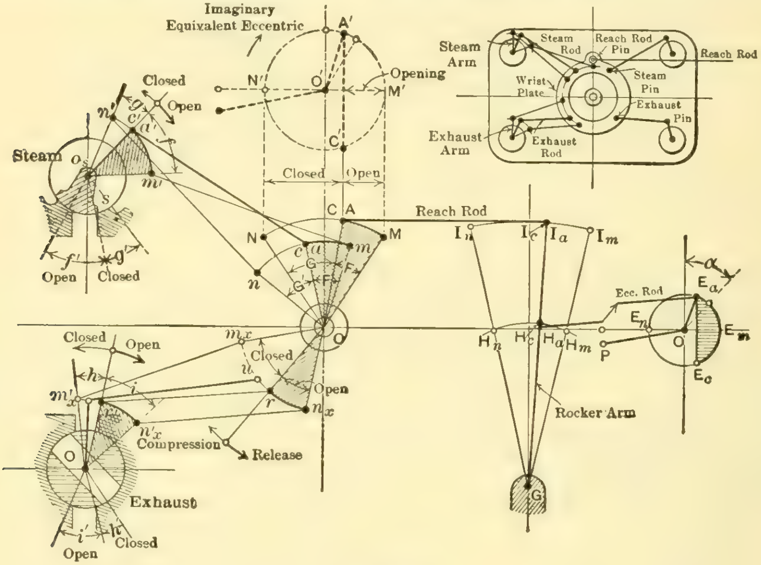


FIG. 446.

The gear shown has, on the side of the cylinder, a wrist plate which the eccentric oscillates through an angle by means of the eccentric rods, rocker arm, and reach rod. In addition to the reach rod pin, the wrist plate has steam and exhaust pins from which steam and exhaust rods lead, respectively, to the steam and exhaust arms, which are keyed to the stems of their valves. The arrangement of these latter parts is such as to give the distorted motion desired.

(f) At the left of Fig. 446, the inner edge *s* of the steam valve is shown even with the port edge with which it opens or closes. Let *g'* be the desired (*small*) angular movement after closure, and f' be the (*large*) angle after opening. The steam arm $o_s a'$, which moves the valve, will swing through the same angles as the valve; thus g and f are, respectively, equal to g' and f'. The pin on the steam arm when in position a' for admission (motion to right) of course coincides with its position c' for cut-off (motion to left), and the angle g for closure is *small* compared with angle f for opening.

Now, starting at the right of the figure, E_a is the eccentric position for admission; E_m, that for maximum opening; E_c, that for cut-off; and E_n is for extreme closure. The similarly subscripted positions of pins H and I on the rocker arm, respectively, correspond with these eccentric positions, as do also the positions A, M, C, and N of the reach-rod pin, and a, m, c, and n of the steam pin. In each case the position for admission (motion to the right) coincides with the position for cut-off (motion to the left). It will be seen that the angular movement $CN(= G)$ of the wrist plate for closure is larger than for $AM(= F)$ opening, which is just contrary to what is desired for the valve movement. However, it is possible by a cut-and-try process to locate the steam pin on the wrist plate in such position, and to use such a length of steam rod, that the steam arm will move through angles $n'c'$ and $a'm'$, respectively, when the steam pin moves through angles nc and am, and thus to accomplish the desired result. With such arrangement the distances mn', aa', and mm' must of course all be equal, since they represent the length of the steam rod.

The exhaust valve motion can be similarly distorted so as to be small after closure and large after release. The arrangement of the steam and exhaust linkages for the crank end of the cylinder is identical with that for the head end, except that it is reversed.

On a high-speed Corliss-valve engine a governor of the shaft type is used, and the cut-off is varied by shifting the eccentric exactly as in the case of the high-speed single-valve engine. The distortion of the motion of the valve in no way affects the timing of events, but merely influences the amount of opening or closure. It would of course be possible to have two wrist plates, one for the steam valves and another for the exhaust, the latter driven by a fixed eccentric so as to obtain constant release and compression.

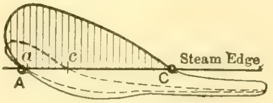


Fig. 447.

There are many other arrangements of linkage used for high-speed Corliss-valve engines, but the underlying principles are much the same as those which have been discussed. In Fig. 447 is given the distorted elliptical diagram for the steam valve of one of these gears.

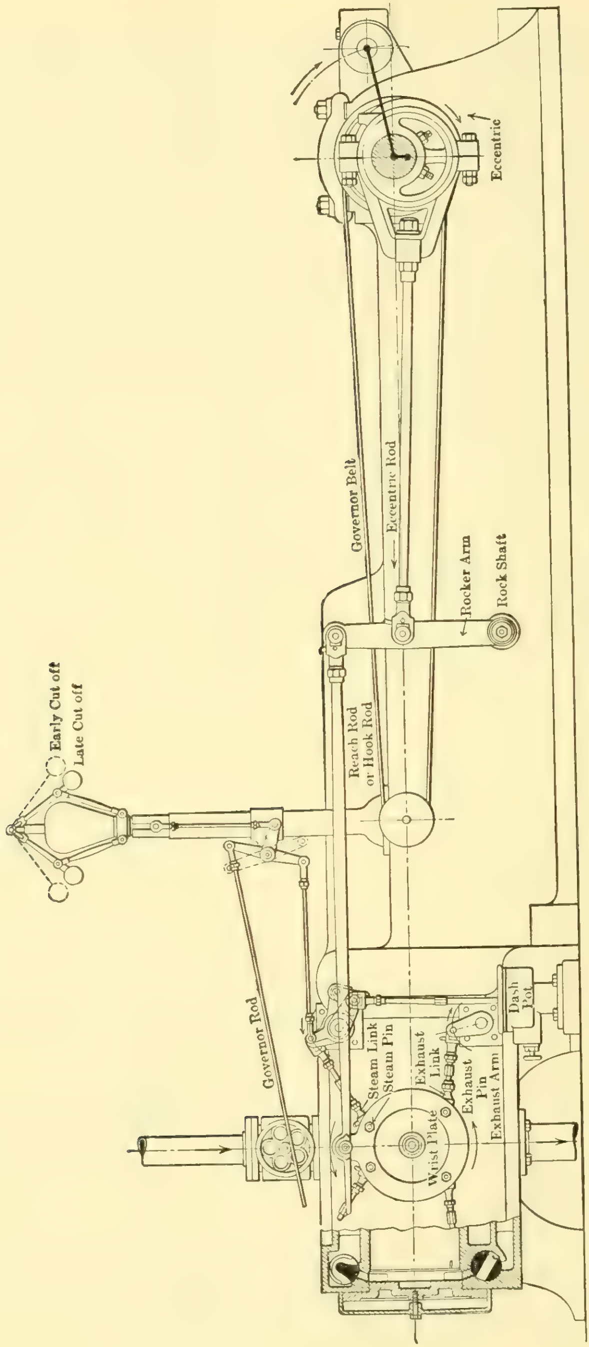


FIG. 448.—Corliss Valve Gear.

(g) **The Trip Cut-off Corliss Engine with Single Eccentric,** Fig. 448, is similar to the arrangement just discussed, except that the eccentric is fixed and the cut-off is operated by a tripping device which is positioned by a governor that is usually of the fly-ball type. The kind of steam valve used is shown in Fig. 449; and the bonnet for the head-end steam valve, and the part of the gear which it supports, are illustrated in Fig. 450 (*a*). The names of these parts are given in Fig. 450 (*b*). The left arm *A* of the bell crank carries a hook *C*, which engages with the rectangular block on the steam arm which is keyed to the valve stem. If the hook remains latched, the motion which the bell-crank arm *B* obtains from the wrist plate by means of the steam rod is transmitted directly to the valve, and the case is identical with that discussed in (**f**) of this section. In these engines, however, as the bell crank is moved clockwise, the hook turns the steam arm and opens the valve only until the part *D* of the hook comes in contact with the *stationary* knock-off cam *E*, when the hook becomes disengaged from the steam arm, which is then returned to its lowest position by the dashpot, thus closing the valve. The governor controls the position of the knock-off cam *E*, which has a definite position corresponding to each different cut-off.

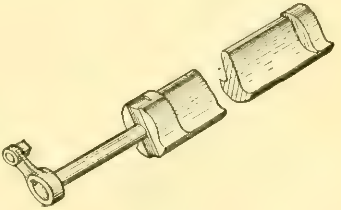
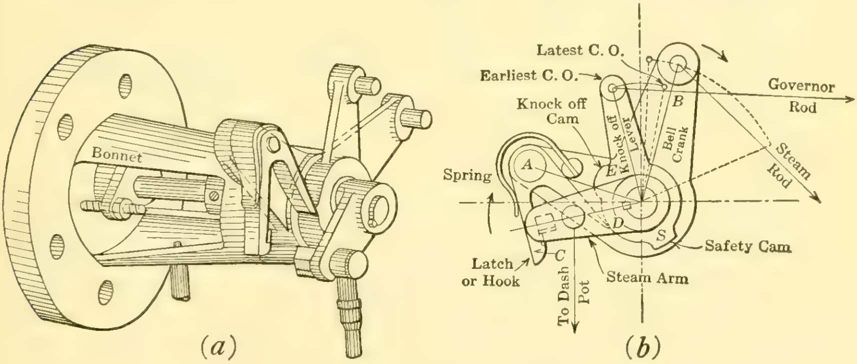


Fig. 449.



(*a*) (*b*)

Fig. 450.—Steam Trip Gear of Corliss Engine.

(**h**) The simple elements of the dashpot are shown in Fig. 451. When the steam valve is opened the plunger is raised and a partial vacuum is formed at *V*. After the hook has been tripped the unbalanced air pressure on the top of the plunger (assisted by gravity) causes the latter to descend rapidly and effect the closure of the steam valve. The fall is stopped by the air cushion which is formed between C and C' and which is adjusted by the cushion valve.

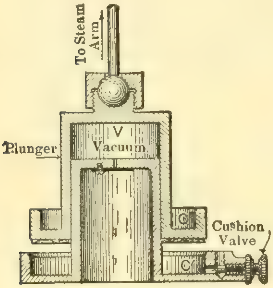


Fig. 451.—Dash Pot.

(i) Fig. 452 is the distorted elliptical diagram for the steam valve. With the trip occurring at t, the cut-off c will occur somewhat later, as the valve cannot close instantaneously. A similar diagram for the exhaust valve is given in Fig. 453.

In connection with Fig. 450, it will be noticed that the trip of the head-end steam valve occurs when the end D of the hook comes in contact with the stationary cam E, while the hook and bell crank are *still moving to the right* (that is, *before the eccentric has reached the right-hand dead center*); and that if, when the eccentric arrives at this dead center, the trip has not taken place (E being too far to the right), it will not occur at all, and cut-off will then be at C' in

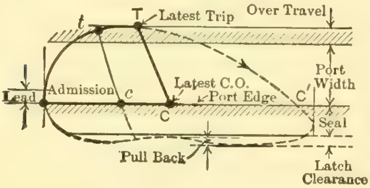


Fig. 452.

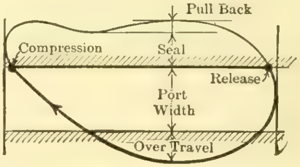


Fig. 453.

Fig. 452. Thus the *latest possible trip* would occur when the eccentric is at the dead center, the valve then having its maximum displacement corresponding to T in Fig. 452. However, to ensure safety in actual operation, the latest trip must always take place before this extreme position is reached, say at t in the same figure.

(j) The angle of advance of the eccentric is fixed by the release and compression, as in the case of the main valve of the riding cut-off gears (Fig. 426). It is in no way dependent on the other events, for, with crank on dead center, the steam rod can be adjusted to give the valve the proper lead, and cut-off is controlled by the knock-off cam independently of the eccentric.

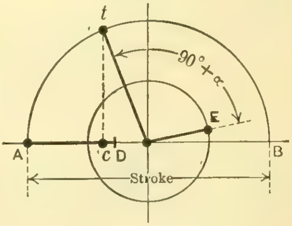


Fig. 454.

If E in Fig. 454 is the eccentric position for latest trip, the crank pin is then at t and the piston at c. As, however, some time must elapse before the valve is closed, cut-off will occur when the piston has reached some position D, which is usually at about 0.4 stroke. Thus with an ordinary *single eccentric Corliss gear the latest cut-off possible is about* 0.4 *stroke,* and this is accomplished only by using the smallest angle of advance that will give the proper release and compression.

(**k**) There are many other arrangements of valves, of trip gear, of wrist-plate linkage, and of dashpot, but all operate in a manner similar to that described. The valves are frequently double- or triple-ported.

(**l**) One of the faults of this gear is that a failure of governor belt stops the governor and lets the weights drop to the lowest position, thus advancing the cut-off to the latest point. The power then developed is greater than that absorbed, and the engine will "run away," and be demolished, unless quickly stopped by hand or by some automatic safety device.

One safety device commonly used consists of a "*safety cam*" S in Fig. 450 which prevents the hook engaging with the steam arm when the governor occupies its lowest position. Some engines have auxiliary governors which will close the throttle valve when the speed becomes unsafe. There are many other forms of safety devices in use.

(**m**) The limitation of the latest cut-off can be avoided by using the **Two-Eccentric Corliss Gear,** in which one eccentric drives a wrist plate for the exhaust valves, and another one actuates the steam wrist plate. Fig. 455 shows the arrangement of eccentrics, with crank on dead center. The angle β between the crank and steam eccentric fixes the latest cut-off. With this arrangement, since the exhaust valves are driven independently, β may be made any value desired (within limits). The angle used in the figure permits of cut-off as late as three-fourths stroke, as is seen from the extreme (dotted) position. In this case the angle of advance of the steam eccentric is negative and equal to $90° - \beta$.

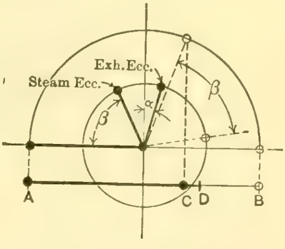


FIG. 455.

(**n**) **Late cut-off** can also be obtained by using a *moving knock-off* cam which may be oscillated by a separate small eccentric, which is at an angle of about 90° with the main eccentric (Nordberg), or by the sidewise motion of the eccentric rod, which is 90° out of phase with the longitudinal movement. With such arrangements, the knock-off cam overtakes the hook and releases it even after the main eccentric has rotated a considerable angle beyond the dead-center position.

(o) As has been seen the rotative speed of trip cut-off Corliss gears must be relatively low, for otherwise the hook gear becomes uncertain of action. Speeds above 150 r.p.m. are seldom used, and generally they are considerably less. Hence engines using this type of gear are commonly classified as "low-speed."

(p) There are several trip cut-off gears which have gridiron valves working across the cylinder either horizontally (somewhat similar in arrangement to Fig. 441) or vertically. Trip cut-off gears are also used with poppet valves (Sect. **331**).

328. Stephenson Link Gear.—(a) The valve gear which until recently has been most commonly used in this country on reversing engines is the **Stephenson Link Gear,** one arrangement of which is shown semi-diagrammatically in Fig. 456. The illustration is for a vertical engine with cylinder above, but the arrangement for a horizontal engine would be identical except for the position of the longitudinal axis.

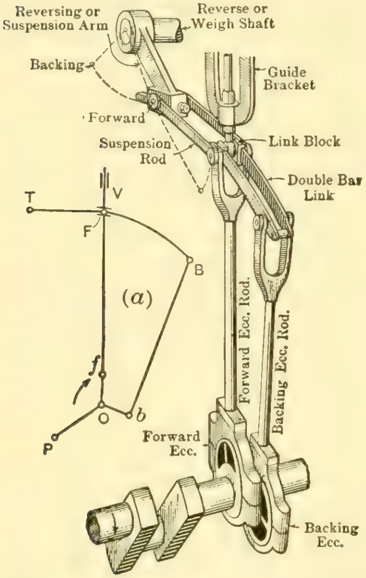


Fig. 456.—Stephenson Link Gear.

The eccentrics are arranged as in Fig. 457, with the *forward eccentric, f,* placed 90° plus angle of advance ahead of the crank in the forward direction of rotation, and the *backing eccentric, b,* at the same angle in the opposite direction. If the valve receives its motion from eccentric *f,* the rotation will be forward (clockwise in this case); and if from *b,* it will be backward (counter-clockwise).

In Fig. 456 the forks at the ends of the two eccentric rods are connected by a *link* (whence the name of this type of gear), different points of which may be brought opposite the *link block* on the end of the valve stem, by turning the *reverse shaft* or weigh shaft. The illustration shows the forward end opposite (in *full gear forward*), hence the valve is receiving all of its motion from the forward eccentric and consequently rotates forward with latest cut-off possible. If the other end of the link is brought opposite the link block (*full gear backing*), the engine would operate backward at maximum cut-off. With the middle of the link opposite (*mid-gear*), the valve receives motion equally from both eccentrics, and will open an amount equal to the lead and close immediately, the cut-off being practically at zero stroke.

(**b**) As the link is shifted from mid-gear toward full-gear forward (or backing) the valve, while receiving motion from both eccentrics, will obtain in increasing amount the major part of its movement from the forward (or backing) eccentric. Accompanying this shifting the width of maximum valve opening is increased, and the cut-off, release and compression are made to occur later, all in a manner quite similar to that which occurs when a gear with single variable eccentric, such as that in Figs. 423 and 424, has the eccentric moved from inner position *3* outward toward *1*. In fact, an **approximate analysis of the Stephenson link gear** can be made by considering the valve as driven by a single swinging eccentric with a radius of path R which can be computed by *McFarlane-Gray's formula*:

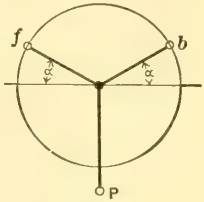


FIG. 457.

$$R = \frac{\text{distance between eccentric centers} \times \text{length of ecc. rod}}{2 \times \text{distance between eccentric-rod pins on the link}} \quad . \quad . \quad (400)$$

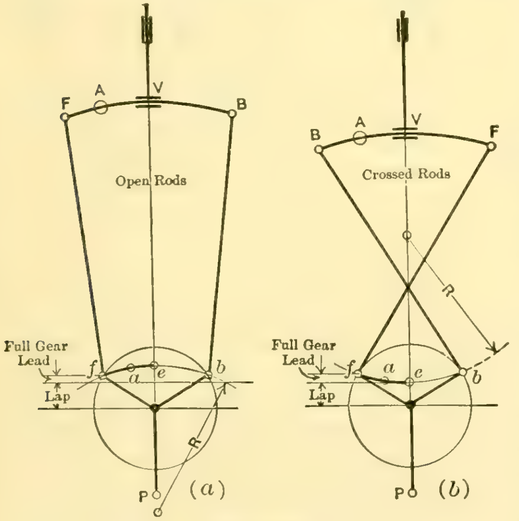


FIG. 458.

(**c**) If, with the crank P pointing *away* from the cylinder, the rods are not crossed, as in Figs. 456 and 458 (*a*), the arrangement is termed an **open-rod linkage.** In this case the path of the equivalent single eccentric is *feb* with radius R. If, with crank in the *same* position, the rods are crossed, as in Fig. 458 (*b*), it is a **cross-rod linkage,** and the path of the equivalent eccentric is *feb*.[8]

For any link position, the equivalent eccentric occupies the same relative position on its path *fb* as the link block does on the link *FB*. For example, if point *A* is held opposite *V*, the motion of the valve resembles closely that which it would have if driven directly by the imaginary eccentric *a* which is so located that $af/fb = AF/FB$. Thus, the approximate action of the valve can be analyzed very simply by constructing any of the usual valve diagrams, provided the throw and angle of advance are taken equal to those of *a*. This analysis is somewhat in error, however, as it neglects the effect of the angularity of the eccentric rods and the slide "*slip*" of the link with respect to the link block. For an accurate determination it is necessary to construct a model or, as in Fig. 459, to draw the mechanism in the various positions that it takes as the crank rotates, and then measure the actual displacements of valve and piston, after which an oval diagram can be constructed to show the true action.

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[8] Note that when the crank has rotated 180° the rods are crossed in the "open-rod" arrangement and open in the "crossed-rod" gear. In classifying the arrangement the crank *must point away from the cylinder.*

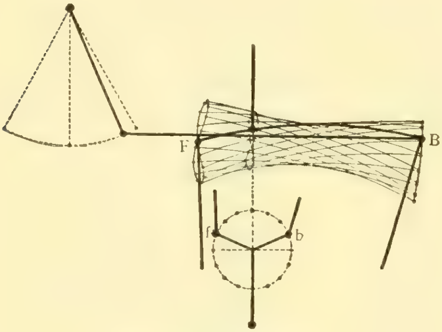


Fig. 459.—Block Slip.

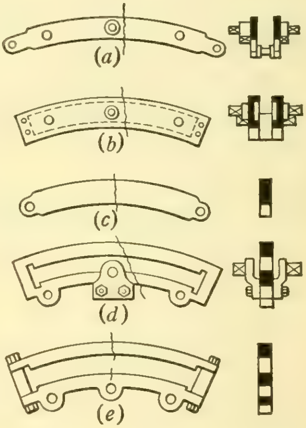


Fig. 460.—Some Forms of Links.

(d) It will be observed from Fig. 458 that the open-rod linkage gives increasing lead as the cut-off is decreased, whereas the reverse occurs with crossed rods; and from Eq. (400) it is seen that using longer eccentric rods increases *R*, thus making the path straighter and the lead less variable. It is also a property of this mechanism that, to have the lead vary equally at the two ends of the valve while shifting from full gear to mid-gear, the radius of the link arc must equal the length *fF* in the arrangement shown in Fig. 458.

(e) The link illustrated in Fig. 456 is of the "double-bar" type. There are many other arrangements: some have the eccentric-rod pins and suspension pins offset from the link arc, as in Fig. 460 (*d*) and (*e*); on some the suspension-rod pin is located at the middle of the link arc, on others it is at the end, and sometimes it is between the middle and the end. The modifications of the valve action introduced in such cases, and the methods of equalizing events, cannot be considered here. For these and certain short cuts that may be used in making an exact analysis of the valve action, see textbooks which are specially devoted to valve gears.

(**f**) When at full gear the link is entirely to one side of the center line of the engine, hence to provide for shifting from latest cut-off forward to that backing a lateral space equal to twice the length of the link is required. On locomotives this gear is placed under the boiler, but on large engines of this kind there is not available sufficient clearance with the roadbed for shifting the link, hence in such cases this type of gear has been displaced by other kinds which require less space or which are more suitably located. In addition to its application on locomotives, the Stephenson link gear has also been widely used on marine and rolling-mill engines, tractors, and on many other types of reversing engines.

329. Other Link Gears.—(**a**) In the **Gooch Link Gear,** Fig. 461, the

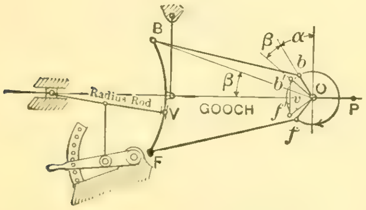


Fig. 461.—Gooch Link Gear.

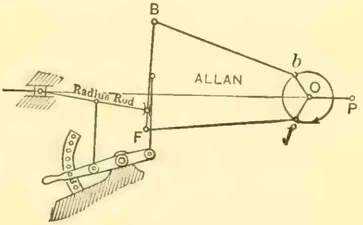


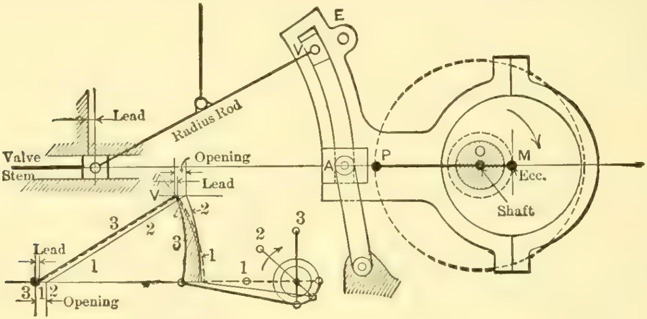
Fig. 462.—Allan Link Gear.

radius rod, instead of the link, is shifted to change the cut-off. As the link radius equals the length of the radius rod, there is no movement of the valve if the adjustment is made when the crank is on dead center, as in the figure; hence the lead is constant. Line $b'f'$ is the path of the equivalent single variable eccentric, $b'bO$ being a right triangle with angle bOb' equal to the angle β between OB and the horizontal axis.

(**b**) The **Allan Link Gear** shown in Fig. 462 has a straight link, and it and the radius rod are shifted in opposite directions in such manner that the valve is not moved when crank is on dead center; hence the lead is also constant in this case, and the path of the equivalent single shifting eccentric is similar to that in the Gooch linkage.

(**c**) The **Porter-Allen Gear** shown in Fig. 463 has a link which is consolidated with the eccentric strap and is guided at A along the center line of the engine. The throw *OM* of the eccentric equals lap plus lead; thus, in the position shown, the head end of the valve is open to lead. As the eccentric rotates from this position the tilting of the link tends to increase the valve opening, but this is opposed by the translatory motion of the link, which tends to close the valve. At (*a*) in the figure, with link block at *V*, maximum opening occurs at crank position *2*, and cut-off at *3*. With link block in a lower position, there would be less opening and earlier closure, the lead remaining the same, however. The radius rod can be raised or lowered either by hand, or automatically by a spindle governor. Motion satisfactory for exhaust valves can be obtained from some such point as *E* on the link.

330. Radial Valve Gears.—(a) In multi-cylinder engines using *link* gears, the valves are usually located between the cylinders and thus add to the length of the engine. In the case of marine engines this necessitates having a greater distance between engine-room bulkheads, which results in reducing the cargo-carrying capacity of the ship. It is true that by using rocker arms the valves might be placed at the sides of the cylinders, but that arrangement of mechanism has certain disadvantages, and even then the space needed on the shaft for the eccentrics prevents, to a certain extent, the shortening of the engine. However, by using the type of valve gears known as "*radial gears*," the valves must necessarily be on the side of the engine instead of between the cylinders; also, most of the gears of this type use but a single eccentric, and in some the eccentrics are dispensed with altogether. With this type of gear the multi-cylinder engine can be made to occupy less space than with link gears; and on locomotives the mechanism may be placed in a more suitable location than is possible with gears of the latter type.



(*a*)

Fig. 463.—Porter-Allen Gear.

There are a great many kinds of radial gears; only the most important will be described.

(**b**) The **Marshall Type of Gear,**[9] which is shown in Fig. 464, uses a single eccentric, either at 0° or 180° with the crank. The point *a* on the eccentric rod *Eab* is guided along path *Gg*,[10] and the end *b* traces the oval figure shown, its positions being numbered to correspond with those of the eccentric and crank. The motion which the valve receives through the radius rod *bc* is practically the same as that obtained from an eccentric. By changing the inclination of the guide *Gg* the oval is modified, the amount of opening is altered and the cut-off is varied. A reversal of the inclination of the guide, as *G′g′*, reverses the engine. The pin *b* may either be located as shown in Fig. 464, or it may be between *E* and *a*, as in Fig. 465. The latter figure also illustrates two methods of guiding the point *a*.

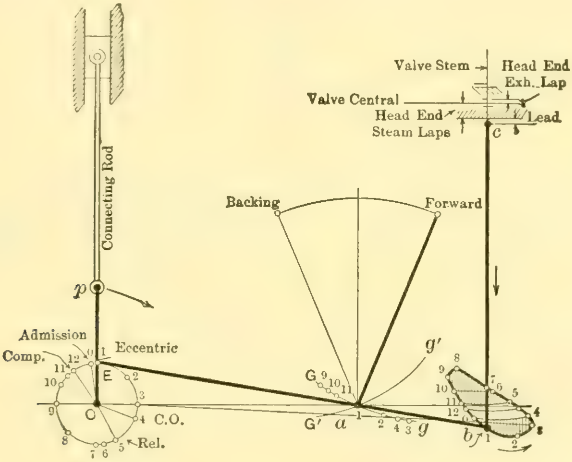


Fig. 464.—Marshall Type of Radial Gear.

(**c**) The **Joy Gear** is somewhat similar to the Marshall, but it dispenses with eccentrics altogether. It makes use of the principle that if any point in a linkage moves in phase with the crank and describes a path that is approximately circular, a pin located at that point can be used as a substitute for the eccentric of the Marshall gear.

The mechanism of the Joy gear is shown in Fig. 466. In this the link *ac* has one end attached to the connecting rod and the other end is suspended by the hanger *fc*. The pin *E* on *ac* takes the place of the eccentric. The rest of the linkage resembles the Marshall in character and performance.

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[9] The names *Klug* and *Bremme* are also associated with this form of mechanism.

[10] The *Hackworth gear* has a straight guide.

The Joy gear has been frequently applied on marine, locomotive, automotive and other engines.

(d) A **composite motion** for driving the valve is used in some radial valve gears, two motions having different phases with respect to the crank being combined for the purpose. Thus, in Fig. 467, if the harmonic motion received from an eccentric H opposite the crank is combined with that from another eccentric V at right angles to the first, the resultant motion is equivalent to that which would be obtained from an eccentric located at E_0 (found by constructing the parallelogram OVE_0H), and a valve receiving this combined motion would operate as if driven by the eccentric E_0. OH is made equal to the lap plus lead, and OV may be varied to change the angle of advance and throw of the equivalent eccentric E_0, which has HE_0 as its path and resembles the single variable eccentric previously discussed. Reversal of the engine may be accomplished by shifting V to the other side of the shaft.

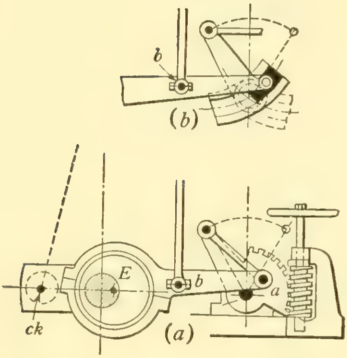

FIG. 465.

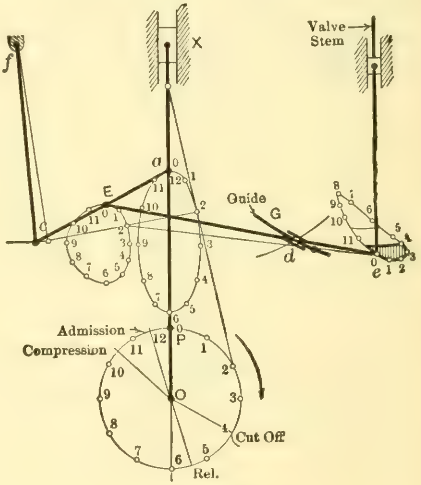

FIG. 466.

(e) **The Walschaert Valve Gear** shown in Fig. 468 uses the principle just considered, but has only one eccentric. The 90° harmonic motion is received from this eccentric, and this motion is combined with one having a 180°-component (or 0°-component, as the case may be) received from the crosshead. The mechanism has a "combination" or "lap-plus-lead" lever ab, one end of which is attached at b to the valve stem and the other end, a, receives motion from the crosshead through a suitable "union" link. To point *c* on this lever is attached a radius rod which connects to a link block *d* in the link *fg*. This link oscillates about a pivot at its middle and receives its motion from eccentric *E*.

If the link block *d* is shifted to *e* at the middle of the link, point *c* will remain practically stationary. Thus, the lever *ab*, considered as pivoted at *c*, can be so proportioned that the end *b* will displace the valve a distance equal to lap-plus-lead when the crosshead reaches the end of its stroke, and will give the valve a motion equivalent to that which it would receive from eccentric *OH* in Fig. 467. With link block shifted to any new position such as *d*, point *c*, and consequently *b*, will receive from the eccentric, *E*, motion which is equivalent to that obtained from eccentric *OV* in Fig. 467. The resultant motion of the valve is that which would be given by the eccentric OE_0 in the latter figure.

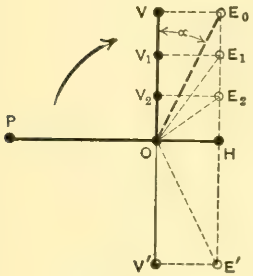

FIG. 467.

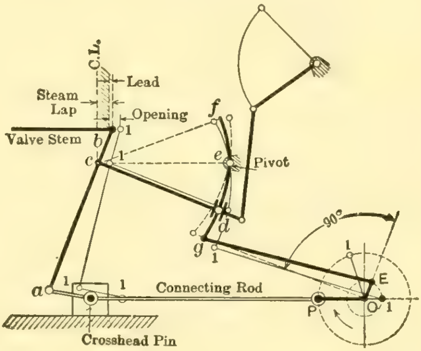

FIG. 468.—Walschaert Radial Valve Gear (with External Valve).

Thus, by shifting the link block *d*, the amplitude of its motion can be varied, and this is accompanied with corresponding changes in the width of valve opening and in the timing of cut-off, release and compression. If shifted to the other side of the pivot *e*, the engine would be reversed.

The Walschaert gear is widely used on locomotives, especially of the larger sizes. Being located on the outer side of the engine, it places no limitation on the size of the boiler, and its parts are readily accessible. Its application to a locomotive is illustrated in Fig. 469.

(f) **The Baker Locomotive Valve Gear,**[11] Fig. 470, is another valve mechanism that employs a composite motion. It has a 90°-eccentric *E* and a lap-plus-lead lever *ab*, which is driven from the crosshead, as in the last gear, but it omits the link. Instead of the latter there is a bar *fgh* which transmits upward motion to *f* when point *g* is guided over the oblique path *GG′*, the end *h* being driven by the 90°-eccentric *E*. The amount of inclination of *GG′* fixes the amplitude of motion of *f*, and the direction of inclination determines whether the engine will operate forward or backward. The vertical motion received by *f* is transmitted horizontally through the bell-crank *fed* and the rod *bd* to point *b*, and, by means of the lap-plus-lead lever, is combined with that received from the crosshead. As shown at (*a*) in the figure, point *g* can be guided by a hanger rod *ig* which is suspended from a yoke that can be turned about trunnions which coincide with the central position of *g*. Thus, by shifting the position of the yoke, the point of suspension *i* and the inclination of path *GG′* can be varied so as to change the cut-off or reverse the engine. It will be noticed that slotted links and sliding link blocks have been avoided in this mechanism and that all parts are joined together merely by pins. In case of wear the pins and their bushings can be replaced readily by new ones.

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[11] Patented (1911). There are also other Baker valve gears. The Baker Traction Engine Gear resembles the Marshall. The Baker-Pilliod Locomotive Gear has been replaced by the simpler one here described.

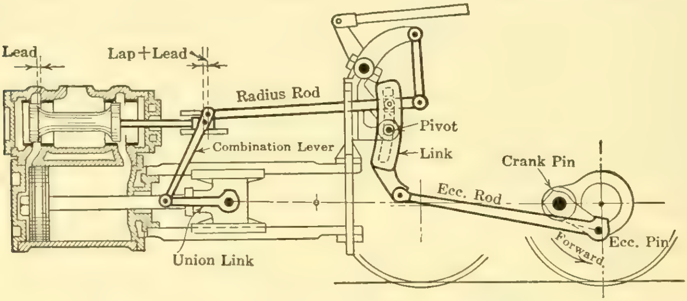

Fig. 469.—Walschaert Locomotive Valve Gear (with Internal Valve).

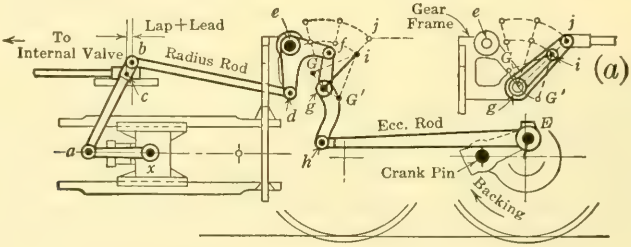

Fig. 470.—Baker Locomotive Valve Gear.

331. Poppet Valves and Their Gears.—(a) As has already been seen, poppet lift-valves function without friction and wear, require no lubrication, and being symmetrical, are not supposed to distort unequally with temperature changes or under high pressures; hence they are suitable for use with high-pressure steam that is highly superheated. Further, they permit the use of short and direct passages between them and the cylinders. The single poppet or mushroom valve, Fig. 471, offers considerable resistance to motion at the time of opening, because of the large unbalanced steam pressure on its back; therefore on steam engines the balanced or partly balanced "*double-beat*" form of valve has commonly been used instead. One form of this double-seated valve is shown in Fig. 472, and is so arranged that the steam pressure on the under side is nearly equal to that above, due allowance being made for the area of the valve stem and for having sufficient inequality of pressure to maintain tightness when the valve is closed. Large valves are commonly placed in *cages*, as in Fig. 472, which fit in suitable bored pockets in the cylinder casting, the two valve decks being joined by posts with steam passages between.

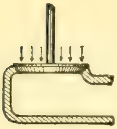

Fig. 471.—Simple Poppet Valve.

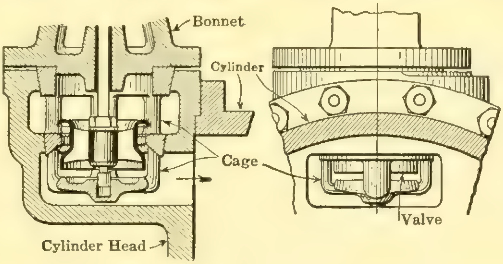

Fig. 472.—Double-Beat Poppet Valve.

As even a very small crack may result in a considerable loss from leakage, it is necessary to make the valves and seats of a double-beat arrangement with great accuracy; and often provision is also made to prevent any slight warping or unequal expansion of the valve and cage, or of other parts affecting them, which might cause uneven seating of the valve faces, or permit one valve edge to close before the other does.[12] Fig. 473 shows several expedients for avoiding such difficulties. Poppet valves are also made in many other forms and are used for exhaust valves as well as for steam.

(**b**) Steam and exhaust valves of this type are commonly operated by cams; and as they function at high speeds, springs or some kind of positive constraint are usually needed to maintain contact between the followers and the cams.

One of the most obvious methods of operating these valves, but least common on steam engines, is to use continuously **rotating cams.** With this arrangement there may be employed a sleeve with a variable cam surface which may be moved endways to change the valve events.

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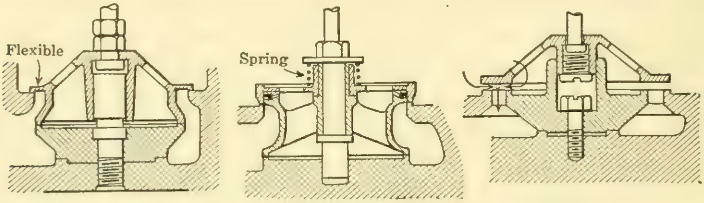
[12] For a more detailed discussion of these difficulties and the methods of overcoming them with single- and double-beat valves, see Stumpf's "The Una-Flow Engine."

The valves may be operated by **reciprocating cams;** but generally **reciprocating rollers** are used, as in Fig. 474; these act on cams that are mounted on the valve plunger, such as *a′a* in the figure. With the latter arrangement, the reciprocating bar can be provided with a reservoir for oil into which the roller dips. This form of gear is often used on small H.S. Uniflow engines (Fig. 366), the reciprocating bar being driven by a shifting eccentric that is controlled by a shaft governor.



(*a*) Stumpf (*b*) Skinner (*c*) Chuse

Fig. 473.

(c) An **oscillating cam,** such as cam "*a*" shown in Fig. 475, may be used, and it may be reciprocated by an eccentric which is shifted by a shaft type of governor to change the cut-off; or it may be driven by a fixed eccentric, or its equivalent, in which case cut-off may be altered by shifting a variable cam, or by changing an intermediate mechanism which may resemble a link or radial gear, or by operating a trip, or by some other means. If the valve is tripped, a dashpot should be used. On large four-valve engines, the eccentrics are commonly mounted, as in Fig. 475, on a lay-shaft, which is parallel to the center line of the cylinder and which is driven from the main shaft by means of miter gears.

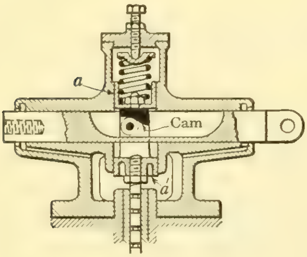


Fig. 474.

(d) The valve may be operated by a single **rolling lever** which fulcrums on a cam surface, such as "*b*" in Fig. 475, and which is driven by an eccentric, or its equivalent. When the valve is being opened or closed the fulcrum point shifts along the surface of the cam; and after closure the lever "floats," being out of contact with the cam, as shown in the dotted position. When used with a steam valve the cut-off can be varied by any of the methods given in **(c)**.

In order to have pure rolling motion the cam surfaces on the lever and on the stationary fulcrum block must be centrodes which are found by determining the successive positions of the instant centers about which the lever and block may be considered as moving with respect to each other, the two ends of the lever having the direction and amount of motion desired.[13]

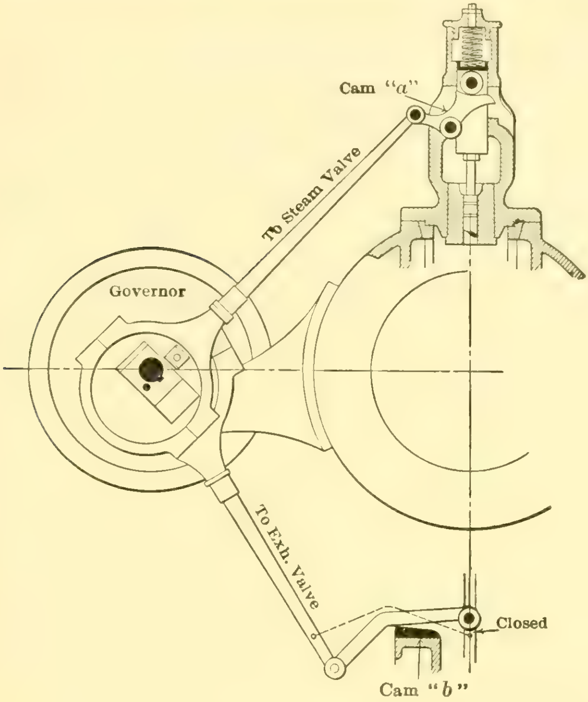


Fig. 475.

(e) The **double rolling lever gear,** Fig. 476, is a valve-operating mechanism that is somewhat similar to the last one except that it employs *two* levers which have cam surfaces that roll together, one lever engaging with the valve stem and the other being driven, as in the foregoing case, by an eccentric or its equivalent. In order to have pure rolling motion between the cam surfaces, their point of contact A must always be on

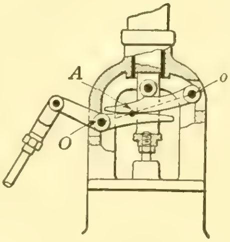


Fig. 476.

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[13] For a discussion of the method of determining these centrodes, see Magg's "Die Steurengen der Verbrennungs-kraftmaschinen," Julius Springer. (Berlin.)

the line of centers *Oo*. In its successive positions this point can be so located that it will divide the distance *Oo* in such proportions as to cause the valve to have motion of the desired character.[14] Rolling lever gears have been used also on large gas engines.

(**f**) Although cams have been discussed here only in connection with poppet valves, they are equally applicable for the operation of piston, gridiron, and other types of valves.[15]

The foregoing treatment, which is necessarily brief and confined to underlying principles, is intended to give the student a general knowledge of the more usual or fundamental types of valves and their gears. There are many other forms, however, but usually they will be found to be modifications or combinations of those here considered.

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[14] *Ibid.*

[15] See List's "Die Steuerungen der Dampfmaschinen" for discussion of many forms of poppet valve (and other) gears.

CHAPTER XXVI

HEAT TRANSMISSION

332. General.—(a) In previous chapters it has been assumed possible to transfer heat from one body to another at will, limited only by the law that a body cannot gain heat from one at a lower temperature unless energy is expended to cause the transfer. It is now necessary to investigate more closely the phenomena connected with the "flow" of heat under the "driving force" of a temperature difference between the two bodies.

(b) The engineering field involving heat transmission is extremely wide and important. Some common examples of heat-transfer apparatus are grouped herewith according to the temperature range covered.

In the *high temperature field*, examples are boilers, water-cooled furnace walls, superheaters, steam reheaters, oil stills, air preheaters, and gas furnace regenerators.

In the *medium temperature field*, there are surface condensers, feed-water heaters, hot-water and steam radiators, economizers, evaporators, automobile radiators, and many others.

In the *low temperature field*, the most common examples are refrigerating coils and similar apparatus, some often used for obtaining much lower temperatures than that needed for ice making.

In general, heat is transmitted by *conduction*, *convection*, or *radiation*, or by some *combination* of them.

(c) Many **symbols** are necessary to express the equations needed in heat transmission, and the *main ones* are given here as a convenient reference and to bring out clearly the distinction between similar ones, such as e, $\mathbf{e}$, and ϵ; additional ones and the significance of special subscripts are also given near the equations involving their use.

A = the surface or cross-section area, in sq. ft.;
C = a constant;
c = the specific heat at constant pressure,[1] in B.t.u. per lb. per deg. fahr.

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[1] The usual subscript p is omitted in this chapter because nearly all engineering applications of heat transmission involve only the constant pressure specific heat, as applied to the hot and cold fluids.

$_c$ = subscript for the cold fluid;
e = the efficiency of a heat-transmitting surface;
$\mathbf{e}$ = the emissivity of a surface;
ϵ = the Napierian base of logarithms = 2.71828;
$\mathbf{h}$ = the film or surface coefficient of heat transmission, in B.t.u. per hr. per deg. fahr. per sq. ft.;
$_h$ = subscript for the hot fluid;
k = the specific thermal conductivity, in B.t.u. per hr. per deg. fahr. per sq. ft. per inch;
q = the rate of heat transfer, in B.t.u. per hr.;
t = temperature, in deg. fahr.;
T = absolute temperature, in deg. fahr.;
U = the overall coefficient of heat transfer, in B.t.u. per hr. per deg. fahr. per sq. ft.;
v = the velocity of fluid flow, in ft. per sec.;
w = the weight rate of fluid flow, in lb. per unit time;
θ = temperature difference, in deg. fahr.;
μ = the absolute viscosity of a fluid, in centipoises;[2]
ρ = the density of a fluid, in lb. per cu. ft.

333. Heat Transmission by Conduction.—(a) Assume the homogeneous metallic bar shown in Fig. 477 to be insulated along its entire length, so that *no heat can be dissipated* by it to the surrounding atmosphere, and also that heat can be continuously supplied to the bar at end A and the same amount be continuously removed from end B. Under these circumstances the heat supplied will all flow along the length of the bar, i.e., flow through the bar. Experience shows that under such conditions the temperature at B will always be lower than the temperature at A, that is, that there must be a *temperature difference* if heat is flowing.

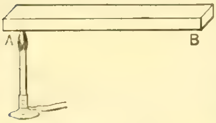

FIG. 477.

This is very much like the phenomena met in the flow of electric currents in similar conductors. It is necessary that a *difference of poten-*

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[2] The *absolute viscosity* of a fluid is defined as the tangential force required to move a plane surface of the fluid of unit area, with unit velocity relative to another parallel plane surface of the fluid at unit distance from the first. In the c.g.s. system, the units of absolute viscosity are dyne seconds per sq. cm., commonly called *poises.* One centipoise = 0.01 poise. Since the absolute viscosity of water at 68 deg. fahr. is one centipoise, the absolute viscosity of a fluid, in centipoises, is also the viscosity of the fluid *relative* to that of water at 68 deg. fahr. In one English system, the units of absolute viscosity are poundal sec. per sq. ft. To convert absolute viscosity in c.g.s. units to these English units, multiply by 0.0672. For example, a fluid having an absolute viscosity of 0.3 poises, 30 centipoises, or 0.02016 poundal sec. per sq. ft., has a viscosity of 30 relative to water at 68 deg. fahr. (See Part III, p. 791.)

tial exist between two points A and B, if an electric current is to flow between them. In the one case, then, electricity flows "because" of a difference of electrical potential or electromotive force; in the other heat flows "because" of a difference of temperature, or, paralleling the above, a difference of "heat potential."

(**b**) Heat conduction is a very complex phenomenon. It is now generally believed that the energy of thermal agitation of the particles of a substance (electrons, ions and atoms) increases with a rise in temperature. Heat conduction therefore consists in transmitting this thermal energy to successive particles along the path of flow. When one end of a solid body is heated the energy of thermal agitation of the first layers of particles is increased; part of this energy is passed on, probably through the agency of electrons, to adjacent layers which thus become heated; and so on through the entire substance.

(**c**) The *general laws* governing the conduction of heat through a body are rather complex because, in general, the flow takes place in several directions, and some heat may be dissipated from several surfaces. The mathematical theory of heat conduction was first propounded by Fourier (1768–1830) but it is beyond the scope of this text. For a strict, mathematical treatment of the **general laws of heat conduction** the student is referred to Ingersoll and Zobel.[3]

(**d**) With the *assumption that heat is flowing only in one direction* the problem is greatly simplified and is still applicable to many engineering cases. Consider the two parallel planes A and B, in the conducting body shown in Fig. 478, to have unit area, to be unit distance apart, and to be maintained at temperatures t_a and t_b, the former being 1 deg. fahr. greater than the latter. Then there will be a flow of heat from A to B; and since no loss from the walls of the intervening body is assumed, heat will have to be supplied at A and removed at B at exactly the same rate as it flows between these points, if these temperatures and the flow are to be maintained constant.

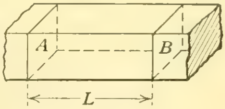

FIG. 478.

Experiment shows that under such conditions a very definite amount of heat will flow from A to B per unit of time in any given material, and this quantity is called the *Specific Heat Conductivity* or, better, the **Specific Thermal Conductivity.** This term will hereafter be designated by the letter k, and, as used in most engineering calculations in this country, it is the number of B.t.u. flowing per *hour* in the material, at right angles to two parallel planes with the area of each equal to *one*

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[3] "Mathematical Theory of Heat Conduction." (Ginn and Co.)

square foot, with *one inch* space between, and with a difference in temperature of *one* deg. fahr. In the c.g.s. system the value of k is expressed in *gram-calories per second,* per deg. cent., per sq. cm. of sections *one* cm. apart. To transpose from the c.g.s. system of units to the other, multiply the value of k by 2903.

(e) The **variation of the specific thermal conductivity** of a conductor with temperature may be rather large, and after considering the actual magnitude of this variation, equations will be developed to show the application of this term. Good conductors of electricity are generally good conductors of heat, and vice versa; also, the thermal and electrical conductivities of metals are extremely sensitive to small changes in composition. No general statement as to the direction of the variation of specific thermal conductivity with increasing temperature can be made, for the conductivity of some materials increases with increased temperature, while for other materials the opposite is true. However, if k_0 represents the conductivity at some datum temperature, the conductivity at any temperature t is given closely by the straight-line law,

$$k_t = k_0(1 + Bt), \quad . \quad (401)$$

in which B is a constant, and may be either positive or negative, depending upon the material.

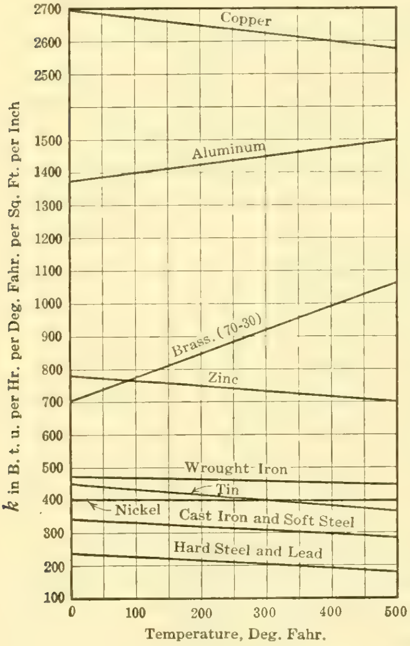

FIG. 479.—Specific Thermal Conductivity of Metals.

Comparison of the specific conductivities of various materials will show that for metals they are several hundred times as great as for water, and that for the latter substance the conductivity is several times the value for gases. Stagnant gases are about the poorest conductors, and stagnant water is nearly as bad.

(f) The **effect of temperature** on the specific thermal conductivity of some **common metals** is shown in Fig. 479. The values used in plotting curves are taken from the work [4] of Lorenz, Jaeger, and Diesselhorst, also of Callendar and Nicolson. The results have been extended beyond the temperatures included in the original investigations, in which the highest was 350 deg. fahr. The agreement in the original work is fair except in the case of cast iron and steel, in which differences in compositions of the metals cause large discrepancies in the results.

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[4] "Heat Transmission by Radiation, Conduction, and Convection," Royds, p. 226.

These discrepancies in the values of k for a *metal*, however, do not seriously affect the solution of the usual heat transmission problem. To understand the reason for this, consider the heat transfer between steam and air occurring when steam flows through a radiator. Experiments show that the resistance to heat transfer *at the two surfaces* of the metal separating the steam and air is much greater than that *through the metal* itself, and whether the radiator is made of cast iron or copper will make little difference in the quantity of heat transferred, other conditions being the same. Consequently, extremely accurate values of the thermal conductivity of a *metal* are usually unnecessary in engineering work.

(g) With **thermal insulating materials,** on the other hand, the internal conductivity of material is of extreme importance. Steam-pipe covering, for example, is usually of sufficient thickness to keep the inner surface of the insulation at a temperature almost equal to that of the steam in the pipe, while the outer surface of the covering will be at nearly ambient temperature. In such a case, the internal resistance of the insulator alone may make up from 75 to 95 per cent of the total resistance to heat transfer, the remaining resistance at the surfaces thus being relatively small.

The specific thermal conductivities of insulators accepted for use as **steam-pipe covering** do not vary greatly. The results of tests reported by R. H. Heilman[5] are shown in Fig. 480, in which values of k are plotted against the arithmetic mean of the outer and inner surface temperatures of the insulation, because the equivalent conductivity of the complete thickness of insulation has been proven to be equal to the conductivity at this mean temperature.[6]

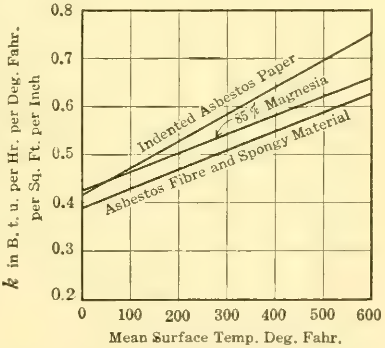

FIG. 480.—Specific Thermal Conductivity of Pipe Covering.

The specific thermal conductivities of some other materials commonly used as **low temperature insulators** are given in Table XXII. The conductivity of most of these insulators

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[5] Mechanical Engineering, Oct., 1924, pp. 593–602.

[6] Trans. A.S.M.E., Vol. 44 (1922), p. 315, and Mechanical Engineering, Oct., 1924, p. 603.

TABLE XXII

THERMAL CONDUCTIVITY OF INSULATORS FOR USE AT LOW TEMPERATURES

Material	Temperature, Degrees Fahr.	Specific Thermal Conductivity k, in B.t.u. per Hour per Degree Fahr. per Square Foot per Inch
Felt	Below 32	0.25
Cork board	32	0.25–0.35
Pure wool	86	0.26
Cork, fine granulated	15–49	0.30
Cork, coarse granulated	10–34	0.32
Cotton	32	0.38
White pine, across grain	86	0.78
Hard rubber	77–122	1.10
Bricks, machine made, dry	32	3.55
Concrete	194	4.44

varies greatly with their density, often increasing as the density increases, due to a reduction of air space. The thermal conductivity of some **refractory materials** is given in Table XXIII. The values given in both of these tables are from the 1924 report of the Am. Soc. Refrig. Eng. Insulation Committee.

(h) The determination of the **thermal conductivity of liquids** has involved much experimental work, a large part of which has been with water. The approximate variation of thermal conductivity of water with temperature is shown in Fig. 481, which gives the average results secured by Weber, Chree, Lees, and Winkelmann.[7] The specific thermal conductivity of olive, castor, and petroleum oils is roughly 30 per cent of that for water.

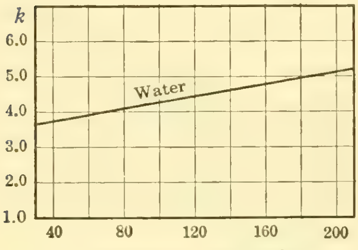

FIG. 481.—Specific Thermal Conductivity of Water.

(i) **The specific conductivity of the common gases,** such as nitrogen, oxygen, ammonia, carbon dioxide, and air, also varies with temperature, the values of k ranging from around 0.15 at 32 deg. fahr. to about 0.2 at 212 deg. fahr. For hydrogen, extremely different values of k are obtained, ranging from 1.0 at 32 deg. to 1.3 at 212 deg.

(j) **The simple laws of heat conduction** have been obtained from

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[7] "Heat Transmission by Radiation, Conduction, and Convection," Royds, p. 229.

TABLE XXIII

THERMAL CONDUCTIVITY OF REFRACTORIES

Material	Temperature, Degrees Fahr.	Specific Thermal Conductivity, k, in B.t.u. per Hour per Degree Fahr. per Square Foot per Inch
Firebrick, average values......	392–2552	6.5–13
Silica brick..................	212–1768	5.8– 9.6
Magnesite brick..............	392–1832	9.3–11.5

experimental observations extending over many years. In general, two important cases need to be considered by engineers. The first one is very simple because it deals with the conduction of the *same amount of heat through equal cross-sectional areas of a body*, such as the flow of heat from one end to another of an insulated metal rod of uniform diameter. The second is slightly more involved because it deals with the *radial conduction* of a constant amount of heat through sections of cylindrical tubes, such as pipe coverings. In the latter case the sections through which the heat flows are progressively becoming larger or smaller, depending upon the direction in which the transfer of heat occurs.

(**k**) **A body of uniform cross-sectional area** and physical properties may be heated until there is a steady flow of heat from the hot end to the cold one. Experiments show that the quantity of heat, q, that can be *conducted* by such a body in a unit time from section 1 to section 2 in a direction perpendicular to these isothermal planes, is directly proportional to k, the specific thermal conductivity of the material, A, the area of the equal cross-sections, $(t_1 - t_2)$, the temperature difference between them, and inversely proportional to L, the distance between them. Therefore, the rate of flow of heat per unit of time is

$$q = kA(t_1 - t_2)/L. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (402)$$

The rate of flow of heat per unit area as well as per unit of time is

$$q/A = k(t_1 - t_2)/L = (t_1 - t_2) \div (L/k) \quad . \quad . \quad . \quad (403)$$

The units for q/A in this country are usually *B.t.u. per hr. per sq. ft.*, as must follow if k is taken in B.t.u. per hr. per deg. fahr. per sq. ft. per inch, L is given in inches, and $(t_1 - t_2)$ in deg. fahr.

When written in the form given, Eq. (403) is conveniently analogous to Ohm's Law for electric circuits with q/A representing current flow rate, $(t_1 - t_2)$, thermal potential, and L/k, *thermal resistance*. This electrical analogy may be carried further by including the case where several plates of different materials are in contact and conducting heat. Here the total **thermal resistance** to conduction is the summation of the individual resistances, or

$$\frac{L}{k} = \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \text{etc.} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (404)$$

For example, consider a plate of cast iron 0.2 in. thick which has a layer of water 0.01 in. thick on one side and a layer of air 0.01 in. thick on the other. Assume that these films are transmitting heat by pure conduction. Suppose the system to be at about 200 deg. fahr. Then, for water, $L_1 = 0.01$, $k_1 = 5$, $L_1/k_1 = 0.002$; for air, $L_2 = 0.01$, $k_2 = 0.2$, $L_2/k_2 = 0.05$; for cast iron, $L_3 = 0.2$, $k_3 = 320$, $L_3/k_3 = 0.0006$; therefore, the total resistance is approximately $= 0.0526$.

(1) A **thick cylindrical tube** of insulating material is very commonly used to retard the flow of heat *to* or *from* a pipe line carrying a fluid that must be kept at a very different temperature from the atmosphere; also relatively thick tubes are used in water-walls of furnaces and other heat-transmitting apparatus. If the thickness of the cylindrical tube wall is greater than about 10 per cent of the inside diameter of the tube, a large error will be involved if Eq. (402) is used. Since most pipe coverings and many metal walls are much thicker than this, a new equation for this case needs to be derived.

Let the inner radius of the tube be r_1, the outer radius r_2, as in Fig. 482. Let the temperature of the inner surface be represented by t_1 and the outer by t_2. Consider the outward flow of a definite quantity of heat, q, per unit of time from the inner surface to the outer surface of the tube having a length l. This same quantity of heat flows across progressively increasing areas of the tube; therefore, for a very thin section at radius r with the area represented by A, the thickness by δr, and the change in temperature across it by $-\delta t$, by applying Eq. (402) to this section,

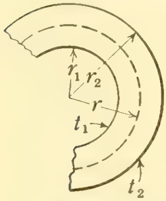

Fig. 482.

$$q = kA\left(\frac{-\delta t}{\delta r}\right) = k2\pi rl\left(\frac{-\delta t}{\delta r}\right). \quad . \quad . \quad . \quad . \quad . \qquad (a)$$

But, as δr becomes smaller and smaller, $-\delta t$ also approaches zero; hence,

$$q = k2\pi rl\left(\frac{-dt}{dr}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (b)$$

Therefore,

$$q\left(\frac{dr}{r}\right) = 2\pi kl(-dt), \quad . \quad . \quad . \quad . \quad . \quad . \qquad (c)$$

and

$$q \int_{r_1}^{r_2} \frac{dr}{r} = 2\pi kl \int_{t_1}^{t_2} - dt, \quad . \quad . \quad . \quad . \quad . \quad (d)$$

therefore,

$$q \cdot \log_\epsilon (r_2/r_1) = 2\pi kl(t_1 - t_2), \quad . \quad . \quad . \quad . \quad . \quad (e)$$

or, with l in ft., r_1 and r_2 in inches, and k based on 1 in. thickness,

$$q = \frac{2\pi kl(t_1 - t_2)}{12 \log_\epsilon (r_2/r_1)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (405)$$

The heat transmitted per hr. per sq. ft. of *outer tube surface*, A_2, is

$$\frac{q}{A_2} = \frac{k(t_1 - t_2)}{r_2 \cdot \log_\epsilon (r_2/r_1)} = \frac{t_1 - t_2}{(r_2/k) \cdot \log_\epsilon (r_2/r_1)}. \quad . \quad . \quad (406)$$

and the heat transmitted per hr. per sq. ft. of *inner tube surface*, A_1, is

$$\frac{q}{A_1} = \frac{r_2}{r_1} \times \frac{q}{A_2} = \frac{t_1 - t_2}{(r_1/k) \cdot \log_\epsilon (r_2/r_1)}. \quad . \quad . \quad . \quad . \quad (407)$$

Note that Eqs. (405), (406), and (407) are also applicable to the flow of heat *inward*, as indicated by the negative results obtained from them when t_1 is less than t_2.

By comparing Eq. (403) with Eq. (406) it will be seen that the two are similar if $r_2 \cdot \log_\epsilon(r_2/r_1)$ be considered as the **equivalent thickness** of a cylindrical tube which has the same thermal resistance as the uniform section.

(**m**) In engineering, the *simple* case of conduction of heat through a *single* homogeneous body is seldom encountered. The flow of heat is usually from some fluid into and through a solid wall and out into a second fluid. The conduction of heat through the solid wall is, then, but one step in a series, and the information given in the preceding sections cannot well be applied to the complete solution of engineering problems until the other methods of heat transfer, which are so intimately tied up with conduction, are considered.

334. Heat Transfer by Convection.—(**a**) When fluids (liquids and gases), have their temperature raised locally, heat may be transferred throughout the mass of the fluid not only by conduction, which depends upon the individual motions of single particles, but also by the mixing of one portion of the fluid with another. The transfer of heat through fluids by this intermixing is called **convection.**

(**b**) There are *two distinct types of tranfer of heat by convection* to be considered.

First, there is the transfer of heat between a surface and fluid, *when* *there is no positive circulation of this fluid produced by mechanical means.* In such a case, the only motion of the fluid is due to **natural or free convection.** One simple example of such heat transfer is illustrated in Fig. 483. Nearly all fluids have their specific volume increased when heated at constant pressure; that is, their density decreases. Local heating will therefore cause local decrease of density, but this will disturb the mechanical equilibrium of the fluid and there will be a tendency for the heated portions to rise. This will be the more marked the more intense and localized the heating; and it results in the upward flow of the heated material which is displaced by the heavier portions, thus causing currents to be formed, or "circulation" to occur. This process very rapidly distributes heat energy to all parts of the mass even though the fluid be a poor conductor of heat. Examples of convection currents caused by local heating are shown by the arrows in Fig. 483.

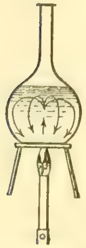

FIG. 483.

Second, there is the transfer of heat between a surface and a fluid when there is an external force causing propulsion of the fluid past the surface, or *when the circulation is forced.* Under such conditions, that portion of the heat transfer due to the intermixing of the fluid is called **forced convection.**

(**c**) When a fluid, even though moving, is in contact with a solid surface, there exists an adhering **fluid film** which may be stationary at the surface and moving at low velocities closely adjacent thereto. The effect of this relatively stationary or stagnant film on the rate of heat transfer by convection was first summarized by **Prof. Osborne Reynolds** in 1874 in a paper presented before the Literary and Philosophical Society of Manchester. In this valuable paper Reynolds says:

> "The heat carried off by air or any fluid from a surface, apart from the effect of radiation, is proportional to the internal diffusion of the fluid at and near the surface, i.e., is proportional to the rate at which particles or molecules pass backwards and forwards from the surface to any given depth within the fluid, for a given difference in temperature between the fluid and the surface.
>
> "The rate of this diffusion has been shown from various considerations to depend on two things:
>
> " 1. The natural internal diffusion of the fluid when at rest.
>
> " 2. The eddies caused by visible motion which mixes the fluid up and continually brings fresh particles into contact with the surface.
>
> "The first of these causes is independent of the velocity of the fluid, and if it be a gas, it is independent of its density, so that it may be said to depend only on the nature of the fluid.
>
> "The second cause, the effect of eddies, arises entirely from the motion of the fluid, and is proportional both to the density of the fluid, if a gas, and the velocity with which it flows past the surface."

(d) *Two distinct types of fluid flow* must be considered before the film theory of heat transfer by convection may be further investigated. One type of fluid flow is called **viscous flow,** in which all particles of the fluid are moving in parallel straight lines. This does not mean that in viscous flow each particle of the fluid must move with the same velocity as every other particle, but only that the parallel stream lines must be maintained. In the case of every fluid, however, if the velocity of flow increases under isothermal conditions, i.e., with no heat supplied nor abstracted (since there is assumed to be no expansion or compression), a certain limiting velocity, known as the **critical velocity,** is attained at which the type of flow suddenly changes from *viscous* to **turbulent flow, which is characterized by motion of the fluid particles in every conceivable direction with innumerable eddy currents.** Prof. Reynolds also suggested that the value of the *critical velocity* for a given fluid depended upon the pipe diameter, D, the absolute viscosity, μ, and the density of the fluid, ρ.

For *isothermal* flow of fluids in pipes, the *critical velocity* is given approximately by the following equations:

$$v_c = (7.6\mu) \div (D\rho) \text{ for iron or steel pipes} \qquad (408)$$

$$v_c = (9.5\mu) \div (D\rho) \text{ for brass or copper tubes,} \qquad (409)$$

where v_c = the critical velocity, in ft. per sec.;
μ = the absolute viscosity of the fluid, in centipoises;[8]
D = the inside diameter of the pipe, in inches;
ρ = the density of the fluid, in lb. per cu. ft.

If heat is being transferred to or from the fluid, the velocity at which viscous flow is superseded by turbulent flow is not given exactly by these equations, because the temperature differences within the fluid set up convection currents.

As an indication of the value of the critical velocity for the isothermal flow of common fluids, Eq. (408) may be solved to give the critical velocity for *air* at 70 deg. fahr. and atmospheric pressure, flowing in a 2-in. (nominal) steel pipe. The equation gives

$$v_c = (7.6 \times 0.0186) \div (2.07 \times 0.075) = 0.9 \text{ ft. per sec.}$$

Since an increase in the pressure of the air does not materially affect its viscosity but does increase its density proportionally, the critical velocity for air at a pressure of 150 lb. per sq. in. abs. and a temperature of 70 deg. fahr., in the same steel pipe, would be approximately $\frac{14.7}{150} \times 0.9$, or 0.09 ft. per sec.

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[8] Defined in Footnote No. 2, p. 252.

For the isothermal flow of *water* at any common pressure and at a temperature of 70 deg. fahr., in a 2-in. (nominal) steel pipe, the critical velocity would be

$$v_c = (7.6 \times 0.99) \div (2.07 \times 62.3) = 0.058 \text{ ft. per sec.}$$

(e) Even though the flow of the fluid at the center of a pipe may be turbulent, there is still a **film** *close to the pipe surface, flowing under viscous conditions,* and the boundary of this film is located at that distance from the pipe surface where viscous flow changes into turbulent flow. It is commonly assumed that the flow of heat from a solid surface to a fluid, or vice versa, is by simple conduction across that part of the fluid flowing in viscous motion, after which the heat is distributed throughout the main body of the fluid by the turbulent mixing. Because of the relatively small resistance offered to heat flow during the turbulent mixing, all of the resistance to heat transfer, and hence the entire temperature drop, is assumed to take place in the viscous film. *Hence the film concept, for all quantitative calculations, reduces heat transfer by convection to heat transfer by conduction through a film of unknown thickness.* If in Eq. (402) the term k/L be replaced by $\mathbf{h}$, which is called the **film coefficient** or **surface coefficient of heat transfer,** then

$$q = \mathbf{h}A(t_1 - t_2), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (410)$$

where q = the rate of heat transfer per unit time, in B.t.u. per hr.;
$\mathbf{h}$ = film coefficient of heat transfer, in B.t.u. per hr. per deg. fahr. per sq. ft.;
A = heat transmitting surface, in sq. ft.;
$t_1 - t_2$ = temperature drop through film, deg. fahr.

The resistance of a fluid film, as represented by $1/\mathbf{h}$, is frequently the largest, and hence the controlling factor, in many engineering problems involving heat transfer.

To illustrate, simply and roughly, consider the transfer of heat by conduction through a copper plate, 0.1 in. thick, for a temperature difference of 1 deg. fahr., with a mean temperature of about 200 deg. fahr. From Eq. (403) and Fig. 479, the rate of heat transfer through the copper would be

$$q/A = k(t_1 - t_2)/L = 2650/0.1 = 26{,}500 \text{ B.t.u. per hr. per sq. ft.}$$

Now if on one side of this copper plate, there is a film of water 0.01 in. thick, and on the other side an air film 0.01 in. thick, the rate of heat transfer by conduction through the two films and plate, for an overall temperature difference of 1 deg. fahr., will be found from Eqs. (403) and (404) to be

$$\frac{q}{A} = \frac{t_1 - t_2}{\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3}} = \frac{1}{\frac{0.01}{5} + \frac{0.1}{2650} + \frac{0.01}{0.2}} = 19 \text{ B.t.u. per hr. per sq. ft.}$$

Note, however, that equivalent to assuming the film thickness, the film coefficient, **h**, might have been assumed. For example, for the water film, the equivalent value of **h** is 500, and for the air film, the equivalent value of **h** is 20 B.t.u. per hr. per deg. fahr. per sq. ft. Whether the assumed thickness of the fluid film (which, with k, determines the value of **h**) is correct, or not, is of little consequence *here* as it is the purpose of this problem merely to illustrate how seriously the rate of heat transfer may be decreased by the presence of such a film. Experimental determination of the rate of heat transfer from water through copper to air would give a numerical result much closer to 19 than to 26,500 B.t.u. per hr. per deg. fahr. per sq. ft., thus showing that the assumption of a stagnant film, or the use of a film coefficient, accounts satisfactorily for the low rates of heat transfer actually found.

The value of this film coefficient for the different cases of convection is necessary before any quantitative problems can be closely solved by this means. The value of **h** for a given fluid depends upon a number of complex variables such as the velocity of fluid flow, the nature and shape of the contact surfaces, and various physical properties of the fluid. Unfortunately, the exact values of **h** for many conditions of heat transfer are unknown at the present time and this is one important reason why any prediction of close quantitative results is impossible in many cases.

335. Natural or Free Convection.—(**a**) The rate of circulation of a **liquid,** and hence the value of the film coefficient of heat transfer for *free convection,* depends principally upon the temperature difference, the viscosity of the liquid, and the shape of the liquid channel. The values of **h** for the different **liquid films** are not very well known, but range from as low as 10, for very viscous liquids with poor convection channels, to 300 for water circulating through free channels.

(**b**) **Gas film resistances** to heat flow are greater than liquid film resistances, as indicated in the numerical problem in Sect. **334** (**e**). The value of the film coefficient for gases will be correspondingly lower than it is for liquids.

One important practical case of transfer of heat through gases by free convection occurs when a hot solid surface is placed in still air; for example, a portion of the transfer of heat from a bare or insulated steam pipe to the surrounding air takes place in this manner. In such cases, however, there is also a transfer of heat by radiation, which may be responsible for a large part of the total loss of heat. This is considered further in Sect. **337.**

Starting with the film theory as a basis, C. W. Rice [9] has deduced by dimensional analysis an equation which gives the rate of heat transfer by free convection. R. H. Heilman [10] has slightly modified this equation to agree with experimental results, and concludes that the loss of heat by *free convection* from surfaces of the principal geometric shapes met with in engineering practice, and placed in *still air*, can be approximated closely by the following equation:

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[9] Trans. A.I.E.E., Vol. 43, 1924, p. 131.

[10] Trans. A.S.M.E., 1929, F. S. P.–51–41, p. 287.

$$\frac{q}{A} = \frac{C\Delta t^{1.266}}{D^{0.2}T_{avg}{}^{0.181}} \quad . \ . \ . \ . \ . \ . \ . \ . \quad (411)$$

where $\frac{q}{A}$ = rate of heat transfer by free convection, in B.t.u. per hr. per sq. ft. of surface;

D = diameter of surface if cylindrical or spherical, or vertical height of surface if plane, in inches;

T_{avg} = average of surface and ambient absolute temperatures deg. fahr.;

Δt = temperature excess of surface over ambient, in deg fahr.;

C = a constant depending upon the surface shape, and having values as given in the following table:

SHAPE	C
Horizontal cylinders (pipes)	1.016
Long vertical cylinders	1.235
Vertical plates	1.394
Horizontal plates facing upward	1.79
Horizontal plates facing downward	0.89
Spheres	1.82

A convenient graphical solution of this equation is given in Fig. 484. Note that the equivalent value of the film coefficient for this case is

$$\mathbf{h} = \frac{C\Delta t^{0.266}}{D^{0.2}T_{avg}{}^{0.181}} \quad . \ . \ . \ . \ . \ . \ . \ . \quad (412)$$

As an illustration of the use of Fig. 484, assume that a horizontal 3 in. bare steel pipe (O.D. = 3.500 in.), at a temperature of 340 deg. fahr., is surrounded by still air at a temperature of 70 deg. fahr. Then $C = 1.016$, $\Delta t = 270$, $t_{avg} = 205$, $D = 3.500$, and from the chart, $q/A = 290$ B.t.u. per hr. per sq. ft.

336. Forced Convection.—(a) Nusselt[11] developed by dimensional analysis the following equation, known as **Nusselt's equation,** as the general expression for the film or *surface coefficient of heat transfer* **for any fluid** *being heated or cooled and flowing inside of a pipe at a velocity greater than the critical.*

$$\left(\frac{\mathbf{h}D}{k}\right) = \phi\left(\frac{Dv\rho}{\mu}\right)\psi\left(\frac{\mu c}{k}\right), \quad . \ . \ . \ . \ . \ . \ . \quad (413)$$

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[11] Mitt. über Forschungsarbeiten, Heft No. 80–96 (1910).

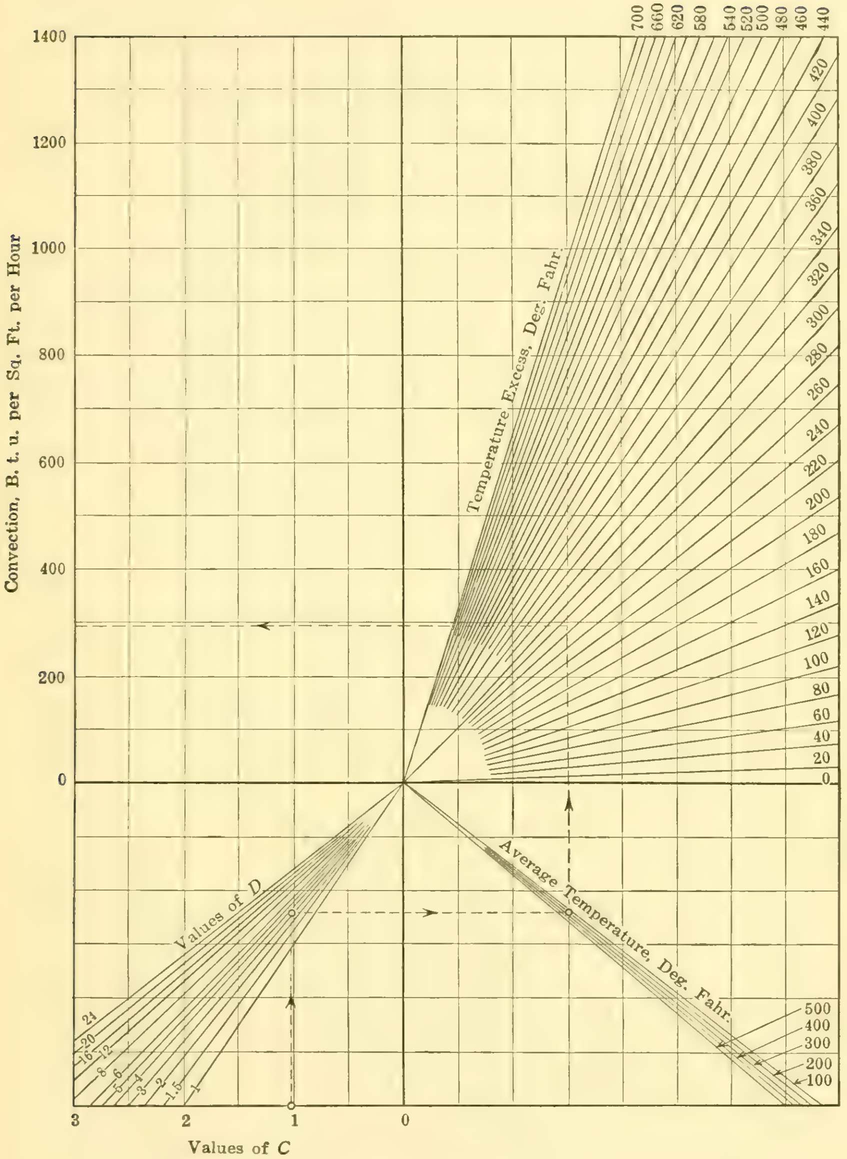

Fig. 484.—Rate of Heat Transfer by Free Convection in Still Air.

where **h** = film coefficient of heat transfer; D = inside diameter of pipe;
k = specific thermal conductivity of fluid; v = velocity of flow;
ρ = density of fluid; μ = absolute viscosity of fluid;
c = specific heat of fluid at constant pressure;
ϕ and ψ = certain functions of the parenthetical quantities.
$\mathbf{h}D/k$ = **Nusselt number,** when dimensionless.
$Dv\rho/\mu$ = **Reynolds number,** when dimensionless.
$\mu c/k$ = **Prandtl number,** when dimensionless.

Note that any consistent set of units may be used, and then each quantity in a parenthesis is dimensionless; also, that the dimensional analysis places no limitations on the functions ϕ and ψ, which must be experimentally determined.[12] Other investigators have sometimes introduced into this equation another dimensionless group, such as the ratio of pipe length to diameter, to express the fact that, due to excessive turbulence produced by changes in section area or by a change in the direction of flow, the rate of heat transfer is locally increased. Many of the recent experimenters start with this general equation and develop special equations to cover the range of their experiments.

(**b**) One of the most recent and most general investigations on the rate of heat transfer for **liquids** *flowing in pipes at velocities greater than critical* is that of Morris and Whitman.[13] Their experiment covered *heating* and *cooling* of **oils** and **water** with a range of velocity from 1 to 20 ft. per second and of viscosity from 0.5 to 55 centipoises. The results of these experiments for the *heating* of the liquids are given in a graphical form in Figs. 485 and 486, in which a hybrid system of units is used. The values of $\phi(Dv\rho/\mu)$ are shown plotted vs. $Dv\rho/\mu$ in Fig. 485. The line was drawn to fair the experimental data. Also the function $\psi(\mu c/k)$ is shown in Fig. 486 plotted vs. $\mu c/k$. Again the line was drawn through the experimental points. The **units to be followed in using Figs. 485 and 486 and Eq. (413) are as follows:** D is in inches, v in ft. per sec., ρ in lb. per cu. ft., μ in centipoises, c in B.t.u. per lb. per deg. fahr., k in B.t.u. per hr. per deg. fahr. per sq. ft. per in., and **h** in B.t.u. per hr. per °F. per sq. ft. With such units, $Dv\rho/\mu$ and $\mu c/k$ are not dimensionless.

These figures may be used to find the *film coefficient of heat transfer for a* **liquid being heated** *inside of a circular pipe with $Dv\rho/\mu$ greater than 5,* in the following manner: First, calculate $Dv\rho/\mu$ and from Fig. 485 find $\phi(Dv\rho/\mu)$. Then calculate $(\mu c/k)$ and from Fig. 486, find

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[12] Nusselt's equation is also written in Exponential Form. See "General Heat-Transfer Formulas for Conduction and Convection," by E. R. Cox, Tr. A.S.M.E., Pet.–50–2, p. 13, 1927–8.

[13] Journal Industrial and Engineering Chemistry, March, 1928, p. 234.

$\psi(\mu c/k)$. Substitute in Eq. (413) the values of the functions found and determine **h**. All physical properties of the liquid used are to *correspond to the temperature of the* **main body** *of the flowing liquid.* The

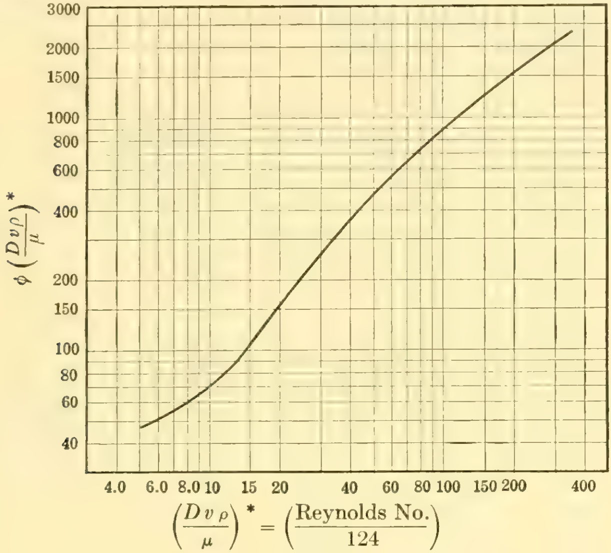

FIG. 485.

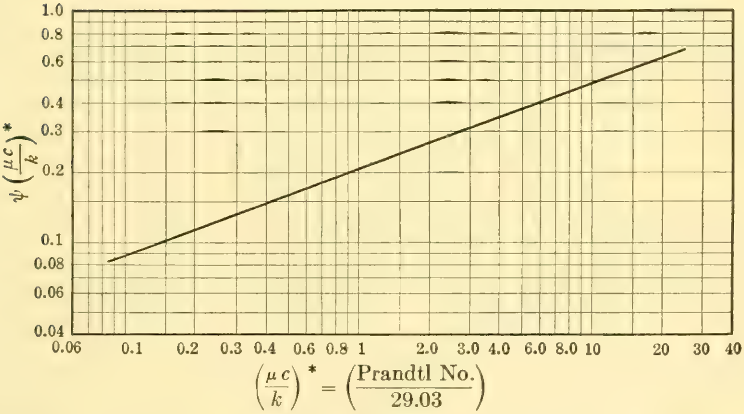

FIG. 486.

experimental values were plotted in this way, and this procedure simplifies the calculations.

As a numerical illustration, suppose that a light crude oil is being heated in a 3-in. (nominal) pipe of a heat exchanger. The velocity of flow is 3 ft. per sec. The crude oil in the pipe is at a temperature of 150 deg. fahr., and at this temperature its viscosity, μ, is 1.5 centipoises, its specific thermal conductivity, k, is 0.95 B.t.u.

* For units, see p. 266.

per deg. fahr. per sq. ft. per inch, its density, ρ, is 50 lb. per cu. ft., and its specific heat, c, is 0.5 B.t.u. per lb. per deg. fahr.; required, the film coefficient of heat transfer, **h**.

First, $Dv\rho/\mu = 3.07 \times 3 \times 50 \div 1.5 = 307$; and, from Fig. 485, $\phi(Dv\rho/\mu) = 2100$. Also $\mu c/k = 1.5 \times 0.5 \div 0.95 = 0.79$; and from Fig. 486, $\psi(\mu c/k) = 0.19$. Then, from Eq. (413), $\mathbf{h}D/k = 2100 \times 0.19 = 399$; hence $\mathbf{h} = 0.95 \times 399/3.07 = 123$ B.t.u. per hr. per deg. fahr. per sq. ft.

From the same experiments, Morris and Whitman also concluded that the film coefficient of heat transfer for a **liquid being cooled** is 75 *per cent* of the value of **h** for the same liquid being *heated* under the same main body conditions. For example, if the crude oil in the preceding problem were being *cooled* in a heat exchanger, instead of being heated, the approximate value of **h** would be 0.75 × 123 or 92 B.t.u. per hr. per deg. fahr. per sq. ft.

Regarding *end effects*, Morris and Whitman used "calming sections" extending 20 pipe diameters on either side of the heating surface to eliminate, as far as possible, the effect of turbulence due to either changes in section or direction of flow. Hence a solution for **h**, using their data, gives the film coefficient for *long pipes*. The effect of end conditions, if important, must be introduced by using a multiplying coefficient, which would be greater than unity.

In case the cross-section of the liquid conduit is not circular, the equivalent value of D may be approximated by (4 × cross-section)/(perimeter of heating surface).

(c) For **heating water** inside *clean* pipes at velocities greater than the critical, McAdams and Frost [14] propose the following equation, which considers the effect of the length of pipe:

$$\mathbf{h} = \frac{138}{D^{0.2}}\left(1 + \frac{50}{r}\right)\left(\frac{v}{\mu}\right)^{0.8}, \quad . \quad . \quad . \quad . \quad . \quad (414)$$

where D = inside diameter of pipe, in inches;
r = ratio of length of pipe to diameter;
μ = viscosity of the water at the mean temperature of the *film*, in centipoises.

h and v are in the same units as before. In case the pipes have been in service long enough to become fouled, the value of **h** will be materially reduced.

This equation is, in form, a contraction of the Nusselt formula, wherein $\rho = 62.3$, $k = 3.95$, and where $\psi(\mu c/k)$ has been eliminated.

To aid in solving Eq. (414) Fig. 487 may be used, as it shows the viscosity of water as a function of temperature. This equation is particularly useful in determining the film coefficient on the water side of *steam* or *ammonia condensers* and *feedwater heaters.*

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[14] Refrigerating Engineering, Vol. 10 (1924), p. 323.

Note that in using Morris and Whitman's curves, the properties of the liquid at the *main body temperature* are employed and that in using the equation suggested by McAdams and Frost, the value of the viscosity of the water, μ, must correspond to the *film temperature.*

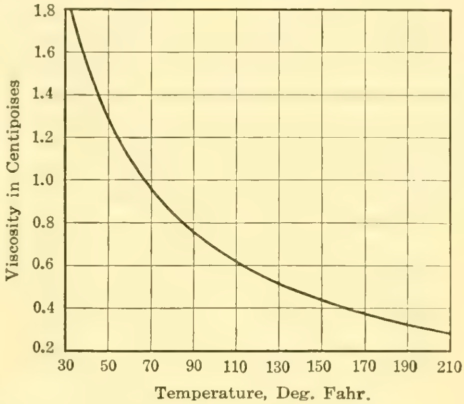

FIG. 487.—Viscosity of Water.

Since Eq. (413) is a general equation for any fluid, and Eq. (414) is recommended for *water,* the two should show some agreement for that liquid. To investigate, assume that water at an *average body temperature* of 56 deg. fahr. is flowing at a velocity of 3 ft. per second through a pipe 65 in. long and 0.902 in. inside diameter and that steam at a temperature of 126 deg. fahr. is being condensed on the outside of the tube; required, the film coefficient of heat transfer, **h**, on the water side. Using Figs. 485 and 486 from Morris and Whitman's data and Eq. (413), **h** is found to be 670 B.t.u. per hr. per deg. fahr. per sq. ft.; using a *film temperature* of 83 deg. fahr. Fig. 487 and Eq. (414) give the same result.

The agreement in this case between Morris and Whitman's data and the equation of McAdams and Frost is striking. However, end effects have been considered by the latter and not by the former. If Eq. (414) be solved for an infinitely long pipe, which would be more consistent with the effect produced by the "calming sections" used by Morris and Whitman, the film coefficient would then be only 400 B.t.u. per hr. per deg. fahr. per sq. ft. Hence it would seem that more experimental work on the effect of end conditions is necessary before any exact prediction of **h** can be made.

However, the equations and data here given for the liquid film coefficient of heat transfer are, in the opinion of the authors, as useful for engineering purposes as any yet presented.

(d) Very little experimental work has been done on the value of the film coefficient of heat transfer with the **fluid velocity less than the critical** *velocity for isothermal flow* as given in Eqs. (408) and (409). The reason for this lack of data is, of course, due to the fact that in most engineering applications, the fluid is forced through the heat exchange apparatus at velocities in excess of the critical. Colburn and Hougen,[15] in studying the rate of heat transmission in tubular gas condensers, derived empirical equations for the *water-film coefficient with vertical flow at velocities less than* 0.1 *ft. per second.* They concluded that if heat be added to a fluid flowing at a velocity lower than the isothermal critical, the original viscous flow of the fluid is altered by convection currents produced by density differences in the fluid stream. They termed this type of flow "**thermal turbulent**" in order to distinguish it from pure viscous flow and from turbulent flow produced by high velocities. This type of flow is peculiar in that an external *force* produces the flow but at velocities lower than those usually associated with *forced convection.* Their empirical equations for the **water-film coefficient** of *heat transfer* for *velocities less than* 0.1 *ft. per second* follow:

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[15] Journal Ind. and Eng. Chem., May, 1930, p. 522.

For *upward flow* in vertical pipes,

$$\mathbf{h} = 0.42\Delta t^{1/3} t_w \quad . \; . \; . \; . \; . \; . \; . \; . \quad (415a)$$

$$\mathbf{h} = 0.37\Delta t^{1/3} t_f. \quad . \; . \; . \; . \; . \; . \; . \; . \quad (415b)$$

For *downward flow* in vertical pipes,

$$\mathbf{h} = 0.49\Delta t^{1/3} t_w. \quad . \; . \; . \; . \; . \; . \; . \; . \quad (416a)$$

$$\mathbf{h} = 0.44\Delta t^{1/3} t_f, \quad . \; . \; . \; . \; . \; . \; . \; . \quad (416b)$$

where $\mathbf{h}$ = film coefficient, in B.t.u. per hr. per deg. fahr., per sq. ft.;
Δt = temperature drop across water film, deg. fahr.;
t_w = temperature of main water stream, deg. fahr.:
t_f = temperature of water film, deg. fahr.

Some work has been done, also, on the rate of heat transfer from metal tubes to water and oil flowing *horizontally* at velocities below the isothermal critical.[16]

(e) Many of the equations commonly used to determine the film coefficient of heat transfer **for gases flowing inside circular pipes** *at velocities in excess of the isothermal critical* are derived from Eq. (413). For *gases,* Nusselt assumed the exponential function of $Dv\rho/\mu$ to be the same as the exponential function of $\mu c/k$ in this equation. He further found the value of this exponent to be 0.786. For *gases,* it follows that $\mathbf{h}D/k$ will be some function of $(Dv\rho c/k)^{0.786}$. Nusselt suggested the following equation for the value of the film coefficient of heat transfer for gases flowing inside pipes at velocities greater than the critical:

$$\mathbf{h} = \frac{16k_p}{D}\left(\frac{Dv\rho c}{k_g}\right)^{0.786}, \quad . \; . \; . \; . \; . \; . \; . \; . \quad (417)$$

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[16] See Dittus, Univ. of California Bulletin 2, No. 11, 1929.

where **h** = gas film coefficient of heat transfer, in B.t.u. per hr. per deg. fahr. per sq. ft.;

D = *inside* diameter of pipe, in inches.;

k_p = specific thermal conductivity of the gas at the pipe temperature, in B.t.u. per hr. per deg. fahr. per sq. ft. per in.;

k_g = specific thermal conductivity of the gas at the temperature of the gas;

v = velocity of flow, in ft. per sec.;

ρ = density of gas, in lb. per cu. ft.;

c = specific heat of gas at constant pressure, in B.t.u. per lb. per deg. fahr.

Noting that $v\rho$ is equal to w/A, which is the *mass velocity* of the gas in lb. per sec. per sq. ft. of cross-sectional area, then

$$\mathbf{h} = \frac{16k_p}{D^{0.214}}\left(\frac{c}{k_g}\cdot\frac{w}{A}\right)^{0.786} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (418)$$

Dixon and McAdams[17] give the following equation for the gas film coefficient for temperatures up to 1200 deg. fahr., with the same units as in Eq. (418):

$$\mathbf{h} = 0.22\frac{cT_f^{0.67}}{D^{0.2}}\left(\frac{w}{A}\right)^{0.8}, \quad . \; . \; . \; . \; . \; . \; . \quad (419)$$

where T_f = the absolute temperature of the gas film, deg. fahr.

It is probable that this equation will also closely give the film coefficient for *superheated steam.* For pipes other than circular, a shape factor equal to (4 × area of cross-section per perimeter of the heating surface) may be substituted for D.

(f) For the **flow of gases outside of pipes** *at velocities higher than the critical,* the available data on the film coefficient of heat transfer are not very complete.

For *air flow at right angles to the length of a single pipe with velocities greater than the critical,* Chappell and McAdams[18] offer the following equation for the film coefficient:

$$\mathbf{h} = \frac{0.8T_f^{1/3}}{D^{0.53}}\left(\frac{w}{A}\right)^{(0.6+0.08\log D)}, \quad . \; . \; . \; . \; . \; . \quad (420)$$

where **h** = the film coefficient of heat transfer, in B.t.u. per hr. per deg. fahr. per sq. ft.;

T_f = arithmetic mean of the absolute temperatures of gas and pipe surface, deg. fahr.;

D = *outside* dia. of pipe, in inches;

w/A = mass velocity, in lb. per sec. per sq. ft. of cross-section.

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[17] "Principles of Chemical Engineering," Walker, Lewis and McAdams (McGraw-Hill).

[18] Trans. A.S.M.E., Vol. 48, 1926, p. 1201.

For air flow at right angles to the length of rows of *equilaterally staggered pipes*, as in transverse tubes of air heaters and economizers, Chappell and McAdams suggest multiplying the value of **h** obtained from Eq. (420), for a single pipe, by the term $(1 + (D/W))$, where W is the distance, in inches, between center lines of rows of adjacent pipes.

(g) The **film coefficients** *of heat transfer* **for condensing vapors** are important in many heat transfer problems. Unfortunately, these values are not closely known for the different vapors. Experiments on **condensing steam** have usually yielded values of the film coefficient between *1500* and *3500* B.t.u. per hr. per deg. fahr. per sq. ft. It is well known that the presence of non-condensable gas in the vapor will materially decrease the value of the film coefficient. In fact, in one experiment,[19] the film coefficient was found to be halved by adding 0.5 per cent of air to air-free steam. McAdams, Sherwood, and Turner [20] found, by using a value of the film coefficient for condensing steam of 2000 and properly combining it with the other resistances to heat transfer, that the observed overall rates of heat transfer in steam surface condensers could be very closely checked. In the absence of more exact information, this value may be used.

In the case of most vapors other than steam, lower film coefficients are shown by experiment. For **condensing ammonia vapor** Sherwood [21] found a value of the film coefficient of 952 B.t.u. per hr. per deg. fahr. per sq. ft. in vertical shell and tube condensers, while Kratz, Macintire, and Gould [22] found a value of the film coefficient of 1635 for condensing ammonia in horizontal shell and tube condensers.

337. Heat Transfer by Radiation.—(a) Experiments show that a system of bodies separated from each other and placed in an insulated enclosure which contains no source of heat will attain an equal temperature; this is believed to be due to a mutual process of radiation and absorption, all of the bodies being supposed to radiate and absorb simultaneously, each to an amount depending upon its constitution, surface-condition, and temperature. When the equilibrium is once attained, radiation is not supposed to cease but continue as actively as before, radiation and absorption being then just equal.[23] When stopped or absorbed by another body or medium, this "radiant energy" becomes evident as heat energy. This does not mean that the radiated energy is in the form of heat when on the way between the two bodies; in fact, if heat energy is to be considered as connected with the motion or condition of molecules, radiant energy of this kind cannot be heat, as radiant energy will pass through a vacuum devoid of molecules of any kind. The name "radiant heat" was formerly given to what is here called *radiant energy*, but in recent years the great increase of knowledge of radiation has made the latter term far more preferable.

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[19] Engineering, June 14, 1929, p. 745.

[20] Trans. A.S.M.E., Vol. 48, 1926, p. 1233.

[21] Refrigerating Engineering, Vol. 13, Feb., 1927, p. 253.

[22] Heat "Transfer in Ammonia Condensers," Part III, Bulletin No. 209, Uni- of Illinois Eng'g. Expt. Sta.

[23] This is known as Prevost's *Theory of Exchanges*. See Prestons *Theory of Heat*.

The exact physical process by which radiant energy is transmitted from one body to another, as for example, from the sun to the earth, is not the matter that chiefly concerns us at the present moment, although it is a subject of great scientific importance and is being constantly studied by physicists. According to one theory, useful in visualizing a possible process, radiant energy is supposed to be transmitted by a vibratory or wave motion in the hypothetical "ether." Whether the ether exists, and whether the process goes on in this way, is really immaterial. The facts remain that a body can lose heat by radiating energy, and that substances can be raised in temperature, vaporized and so on, by receiving such radiated energy.

The known spectrum of "ether waves" comprises a very great range of wave lengths extending from the longest radio waves, whose lengths are measured in miles, to the very short gamma rays of radium whose wave lengths are of the order of $4(10)^{-11}$ inch. The term *radiant energy*, strictly speaking, applies to this entire spectrum. Whenever a body absorbs any of this radiant energy, heat or some other form of energy appears. The *wave lengths* of the *visible* part of the spectrum extend from about $4(10)^{-5}$ cm. (violet) to about $8(10)^{-5}$ cm. (red).

(**b**) The **wave lengths** mainly involved in engineering applications of **heat transmission** are known to be longer than those of light, and it is to be especially noted that this constitutes their only difference.

As the temperature of a radiating body increases, a greater portion of its energy leaves as light waves, but even at the high temperature of an electric light filament, say 4000 deg. fahr., only about 4 per cent of the energy is radiated as light waves. At extremely high temperatures, such as that of the sun's surface (about 11,000° F.), light waves constitute a much larger portion of the radiated energy.

(**c**) Substances differ in their radiating and absorbing capacity, and the "*black body*" is used as a standard for reference. The **ideal black body** is defined as one which absorbs all radiation falling upon it. In other words, this body has an *absorptivity*, a, of unity. The **total emissive power**, E_B, of a perfect black body is the time rate at which it emits radiant energy per unit area, and is independent of the material of the body, depending only on its temperature. For other bodies, the *total* *emissive power*, E_s, depends also on the kind of surface; E_s is given in B.t.u. per hr. per sq. ft. In 1879, Stefan first suggested that the total radiation from a body is proportional to the fourth power of the absolute temperature. In 1894, Boltzmann proved this law to hold good for black body radiations; and the **Stefan-Boltzmann Law** becomes [24]

$$q_B = 17.23(10)^{-10} A_B T_B{}^4 \quad . \quad . \quad . \quad . \quad . \quad (421)$$

where q_B = the total energy emitted from the black body, in B.t.u. per hr.;
A_B = the surface area of the black body, in sq. ft.;
T_B = the absolute temperature of the black body, in deg. fahr.

There is no perfect black body, but it has been demonstrated that the radiation from a small hole in an opaque enclosure at a uniform temperature fulfills very closely black body conditions, independent of the substance forming the enclosure, and, further, that any substance in such an enclosure behaves as a black body.

(d) **Actual surfaces** behave differently from the ideal black body in that they absorb only a portion of the incident radiation and reflect some. A body is said to be **grey** when in each wave length it radiates but a fraction of the energy radiated by a black body at the same temperature, this fraction being *constant* for all wave lengths.

Most bodies, however, are selective in their radiation and absorption, i.e., in each wave length their absorptions and radiations do not bear a constant proportion to the absorptions and radiations of a black body at the same temperature; hence they are neither " black " nor " grey."

At a given temperature, the quotient obtained by dividing the total emissive power by the absorptivity of *any* body *is the same for all bodies.* Thus, this ratio of emissive power to absorptivity is the same for any actual body as it is for the ideal black body at the same temperature, and since the absorptivity of the black body is unity, this quotient is equal to the total emissive power of a black body at the same temperature. This is known as **Kirchhoff's Law.**

The **emissivity, e,** of a body is defined as the ratio of its total emissive power, E_s, to the total emissive power of a black body, E_B, at the same temperature. From Kirchhoff's Law, it follows that the emissivity of a body is equal to the absorptivity and each is a simple numerical ratio without units. The emissivity of a body can never be greater than unity, for the emissive power of a black body is the maximum possible.

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[24] See "Surface Heat Transmission," by R. H. Heilman, Trans. A.S.M.E., 1929, F.S.P.–51–41, p. 287. Also W. W. Coblentz, Bureau of Standards Technical Paper No. 406.

In general, rough, black, or sooted surfaces have greater emissivities and greater absorptivities than polished surfaces. Approximate values of the emissivity, $\mathbf{e}$, of various surfaces for temperatures below 2000 deg. fahr. are given in the following table:

Lampblack, rough steel plate, rough and oxidized iron or steel	0.95
Brick or asbestos paper	0.90
Oxidized copper or brass	0.60
Polished iron, or steel, or unpolished brass	0.20
Polished copper, brass, or tin	0.10

(e) As any radiating body is surrounded by other radiating bodies, it must be *receiving* as well as *emitting* radiant energy. Hence, the net result of such an interchange would be a loss or gain of energy equal to the difference between that radiated and that absorbed. **For two parallel plane surfaces** (*of any materials*) *infinite in extent, at absolute temperatures of T_1 and T_2 deg. fahr., respectively, and separated by a vacuous space, the net heat lost from the hot surface is given by the following equation, which is a* **deduction from** *the* **Stefan-Boltzmann Law:**

$$q = \frac{17.23A(T_1^4 - T_2^4)}{(10)^{10}\left(\frac{1}{\mathbf{e}_1} + \frac{1}{\mathbf{e}_2} - 1\right)} = \frac{1723A}{\frac{1}{\mathbf{e}_1} + \frac{1}{\mathbf{e}_2} - 1}\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4\right], \quad (422)$$

where q = net radiation from hot surface, or energy interchanged, in B.t.u. per hr.;
A = the area of either surface in sq. ft.;
$\mathbf{e}_1$ = emissivity of hot surface;
$\mathbf{e}_2$ = emissivity of cold surface;
T_1 = absolute temperature of hot surface, deg. fahr.;
T_2 = absolute temperature of cold surface, deg. fahr.

Eq. (422) also applies to the case where *one body is entirely enclosed in another body if the two are of nearly equal size,* as, for example, two concentric spheres of nearly equal diameter; the surface area of the *enclosed body* should be substituted for A.

If the enclosed body is very small *relative to the enclosure,* the inner walls of the enclosure behave as black bodies, irrespective of their character and therefore $\mathbf{e}_2 = 1$. Then, for this case, Eq. (422) becomes

$$q = 1723\,\mathbf{e}_1 A_1\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4\right], \quad . \quad . \quad . \quad (423)$$

where $\mathbf{e}_1$ = emissivity of the *enclosed surface*;
A_1 = the area of the *enclosed surface,* in sq. ft.

If both surfaces are black, $\mathbf{e}_1 = \mathbf{e}_2 = 1$: and for parallel planes of infinite extent or for one body enclosed in the other, Eq. (422) becomes

$$q = 1723A\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4\right]. \quad . \quad . \quad . \quad . \qquad (424)$$

(f) Radiant energy, like light, travels or radiates *in all directions* from the surface of the body which serves as its source, and since radiating and receiving surfaces are not always arranged so that all energy lost by one is received by the other, there are many complications in calculating the energy interchange. Detailed information as to how each individual case is handled will not be given here, but some of the fundamentals will be explained.

Referring to Fig. 488, δA is a very small radiating surface, and δB is a minute portion of the receiving surface. The line ON is normal to the area δA; the center to center line OR makes the angle θ with this normal; the area δB subtends at O the solid angle $\delta\phi$. The assumption may be made that the rate at which a surface radiates energy in a direction θ from the normal to the surface is proportional to cos θ, (and this is approximately true). Then, the rate at which energy is interchanged by radiation between the two surfaces depends upon the character and temperature of each surface, and is proportional to the solid angle, $\delta\phi$, and also to cos θ. One common way of revising Eq. (422) to cover cases of finite surface is as follows:

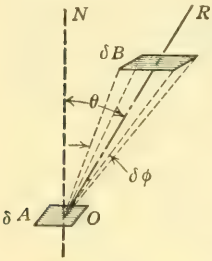

Fig. 488.

$$q = 1723AF_aF_e\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4\right], \quad . \quad . \quad . \qquad (425)$$

where A = the area of one of the two surfaces in sq. ft.;
F_a = an *angle factor* to allow for incomplete reception of all radiant energy emitted. It depends on the average solid angle throughout which one surface "sees" the other, upon the angle θ, and also upon which surface is chosen to use in the area term, A;
F_e = an *emissivity factor* to allow for surfaces other than black bodies, and depends in value upon the emissivities of the two surfaces.

In order to evaluate the angle factor, F_a, for any given arrangement of the surfaces, the value of $\int \cos\theta \cdot d\theta$ must be determined. This is commonly done either graphically or mechanically. One important arrangement of surfaces, for which Hottel[25] gives values of F_a and F_e, is shown in Fig. 489. In this case energy is being radiated from the lower plane surface to rows of tubes on equilateral triangle centers.

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[25] "Radiant Heat Transmission," by H. C. Hottel, Mechanical Engineering, July, 1930, p. 699.

The curve shows the angle factor, F_a, as a function of the ratio of the tube center to center distance to the tube diameter. Curve (*3*) gives the direct radiation to the first row; curve (*5*) gives the total radiation to the first row when a non-conducting but reradiating surface is placed above the tubes, and when there is only one row of tubes. For two tube rows, curve (*1*) gives the direct radiation to the second row; curve (*2*) gives the total radiation to the second row; curve (*4*) the total radiation to the first row, and curve (*6*) the total radiation to the two rows when the reradiating surface is present. In this case, the value of the angle factor, F_a, is based upon using for A in Eq. (425) the area of a plane replacing the tubes; F_e is equal to $\mathbf{e}_1\mathbf{e}_2$, the product of the emissivities of the radiating and receiving surfaces. Fig. 489 shows that, for a ratio of the center to center distance to the tube diameter of 2.5, adding the reradiating surface when there is only one row of tubes increases the angle factor, F_a, from 0.55 to 0.8; while adding the second row of tubes, and retaining the reradiating surface, increases the angle factor from 0.8 to 0.96.

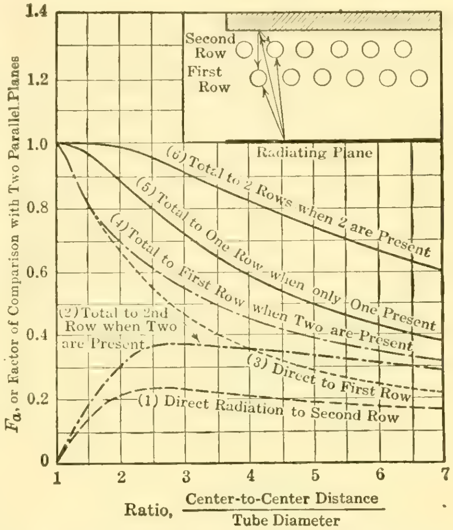

FIG. 489.—Radiation from a Plane to One or Two Rows of Tubes above and Parallel to the Plane.

(Tubes on equillateral triangular centers; non-conducting but reradiating refractory surface above tubes. Ordinate expressed on basis of heat transferred from a plane to a plane replacing the tubes, or to an infinite number of rows of tubes.)

(g) The **radiation from non-luminous gases** *does not obey the radiation law for solids* as previously given in Eq. (422). Of the gaseous products of combustion, oxygen and nitrogen emit, and therefore absorb, very little radiant energy. Water vapor, carbon dioxide, carbon monoxide, the hydrocarbons, and sulphur dioxide all emit and absorb radiant energy to an appreciable degree. However, carbon dioxide and water vapor are the only gases which are commonly present in products of combustion in sufficient quantities to emit much energy. At low temperatures and with thin gas layers, the energy radiated by these gases may be negligible, but for high temperatures and thick gas layers, the energy radiated or absorbed by these gases may be a very appreciable portion of the total radiation.

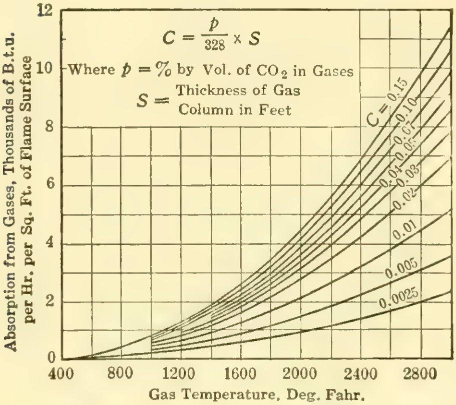

FIG. 490.—Cold Surface Absorption from Carbon Dioxide.

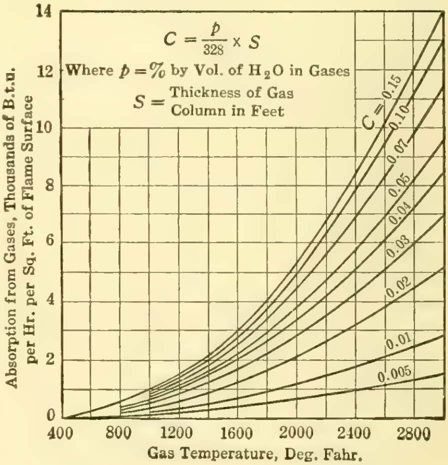

FIG. 491.—Cold Surface Absorption from Water Vapor.

Radiation from non-luminous gases is independent of gas velocity, but varies with the size of the heat transfer apparatus and with temperature. The radiation from gases is difficult to measure for the following reasons: (1) Coatings of slag and oxide on metals affect the absorbing power of these metals to a very marked degree; (2) the temperature of the gases in a furnace is extremely hard to measure and yet the rate of radiation is largely dependent upon this factor; (3) the composition and thickness of the gas layers under investigation affect the radiation; (4) conduction and convection are likely to affect the measurements; (5) the products of combustion may carry very small particles of solid carbon, and the fineness and amount of this solid matter in suspension will affect the radiation from the gases. In Figs. 490 and 491 the respective rates of heat absorption from carbon dioxide and water vapor by a metal-absorbing surface are shown. The curves are given as computed by Wohlenberg and Lindseth[26] from Schack's[27] equation for an absorbing surface with an absorptivity of 0.95, at a surface temperature of 900 deg. fahr. abs., which corresponds to a boiler pressure of 365 lb. per sq. in. gage in a water-cooled furnace. However, the application of these curves anywhere in the range of boiler pressures from 100 to 600 lb. per sq. in. for furnace temperatures of 2000 deg. fahr.

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[26] Trans. A.S.M.E., 1926, pp. 923 and 924.

[27] "Radiation in Boiler Furnaces," B. N. Broido, Trans. A. S.M.E.,1925, p. 1146.

or higher, is recommended as giving sufficient accuracy in the light of present knowledge and the necessary assumptions.

These curves show that the radiation from gases increases with temperature and with the thickness of the gas column, up to a so-called critical value of the function C which is about 0.15. Thus for example, the maximum radiation from a gas containing 20 per cent CO_2 by volume would be obtained when the gas column had the thickness,

$$S = \frac{328C}{\%\,CO_2} = \frac{328 \times .15}{20} = 2.46 \text{ ft.}$$

(**h**) **Radiation from the suspended matter** *in powdered coal flames* **and from luminous gases** is frequently of a greater order of magnitude than radiation from non-luminous gases. Although quantitative experimental data are still very meager, the method usually followed is to assume that radiation from the suspended solid material follows the general radiation law, as given in Eq. (422). If the emissivity of the solid matter in the flame can be calculated from the known variables upon which its value depends, Eq. (425) may be applied.

The suspended solid material in powdered coal flames consists of particles grading in composition from pure carbon to nearly pure ash and with an average size of perhaps 0.001 inch. Haslam and Hottell[28] have developed a rational method of determining the emissivity of this suspended solid material. The various factors which they have included as affecting this emissivity are the effective flame thickness, the flame temperature, the weight ratio of combustion products to fuel, the density of the coal and of the coked product, the initial particle size of the coal, and the fractions of volatile matter and moisture in the coal. They conclude that the value of the emissivity of the particles suspended in powdered coal flames may be expressed by an equation of the following form:

$$\mathbf{e} = 1 - (0.368)^{\frac{nS}{T}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (426)$$

where S = the effective flame thickness;
T = the absolute flame temperature;
and n depends upon the other variables mentioned above.

In **luminous flames,** the suspended matter consists of carbon or soot of a much smaller particle size than the solids in powdered coal flames. This carbon results from the "cracking" of the hydrocarbons. For high flame temperatures and small differences between the true flame temperature, as measured with a high velocity thermocouple, and the brightness temperature, as determined with an optical pyrometer, the value of the *emissivity* is very high, approaching unity as a limit. Dr. Lent [29] has added benzene to form soot in blast furnace gas flames, with resulting radiation from the flame approaching that of a perfect black body at the same temperature. Haslam and Boyer [30] found that the heat received from a luminous acetylene flame, with an emissivity less than unity, was approximately four times as much as from a corresponding non-luminous flame.

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[28] Trans. A.S.M.E., 1927–1928, F.S.P.–50–3, p. 9.

(i) **Several numerical examples** will now be given to illustrate the use of the preceding principles and equations.

1. Consider the determination of the heat transfer by radiation in a boiler furnace. In a stoker-fired furnace, there will be radiation from the fuel bed and from the gaseous products of combustion. In a pulverized coal furnace, there will be radiation from the solid particles in suspension and from the hot gases. In each case, of course, there will be a transfer of heat by convection as the hot gases sweep over the cold surfaces. Assume a pulverized coal furnace of dimensions 20 ft. × 20 ft. × 20 ft. with flame dimensions of 19 ft. × 19 ft. × 19 ft.; flame temperature of 2500 deg. fahr.; cold surface temperature of 440 deg. fahr., and absorptivity or emissivity of 0.95; CO_2, 14.2 per cent by volume, and H_2O vapor, 6.2 per cent by volume in the products of combustion.

Equation (426) gives the emissivity of the solid particles in a pulverized coal flame. In this equation, the value of n is influenced by the size, density, and composition of the coal, and the weight ratio of products of combustion to fuel. For this case, the value of n may be taken as 90. The effective flame thickness may be taken as two-thirds of the actual thickness, or about 12.5 ft.[31] Then, for the solid particles,

$$\mathbf{e} = 1 - (0.368)^{\frac{90 \times 12.5}{2960}} = 1 - (0.368)^{0.38} = 0.31.$$

There are six flame surfaces of an area of 361 sq. ft. each, parallel to exposed cold surfaces. Thus, the heat absorbed by the cold surfaces from the solid particles of the flame is closely given by Eq. (422) as,

$$q = \frac{6 \times 361 \times 1723}{\dfrac{1}{0.31} + \dfrac{1}{0.95} - 1}(2.96^4 - 0.9^4)$$

$$= 86{,}000{,}000 \text{ B.t.u. per hr.}$$

The radiation from the CO_2 and H_2O vapor in the products of combustion may be found from Figs. 490 and 491. For the CO_2, the value of the critical thickness would be, $(328 \times 0.15) \div 14.2 = 3.46$ ft. The actual flame thickness exceeds this value, but the maximum possible radiation is secured when $C = 0.15$. Using this value, the heat absorbed by the cold surface from the CO_2 at 2500 deg. fahr. is given by Fig. 490 as 7500 B.t.u. per hr. per sq. ft. Similarly, from Fig. 491, the heat absorbed by the cold surface from the H_2O vapor is given as 9300 B.t.u. per hr. per sq. ft. However, there is interference between the radiation from the solid particles and from emissions of the gases in the same wave length regions. This produces a "shielding" effect, which has been analyzed quantitatively by Wohlenberg and Lindseth.[32]

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[29] Abstract of original article, in Mechanical Engineering, June, 1926, p. 610.

[30] Ind. and Eng. Chem., Vol. 19, No. 4, 1927.

[31] See Trans. A.S.M.E., 1927–1928, F.S.P.–50–3, p. 11.

For the case under discussion, if a "shielding" coefficient of 0.86 be used, the heat absorbed by the cold surface from the gases is

$$q = 0.86 \times 6 \times 361 \times (7500 + 9300) = 31{,}200{,}000 \text{ B.t.u. per hr.}$$

Then, in this example, the cold surface would absorb about 117,200,000 B.t.u. per hr., due to the combined radiation from the solid suspended particles and the gaseous products of combustion. This absorption rate amounts to 14,650 B.t.u. per hr. per cu. ft. of furnace volume, and, assuming that all six walls of the furnace absorb the heat equally, the rate of heat transfer per sq. ft. of this surface is 48,800 B.t.u. per hr.

2. Another illustration of the quantitative use of the foregoing principles will be given. From Eq. (411) the heat loss by free convection from a 1-in. bare steel pipe, at a temperature of 500 deg. fahr. in still air at a temperature of 80 deg. fahr., amounts to 610 B.t.u. per hr. per sq. ft. of pipe surface. Suppose that the *total loss* of heat from the pipe is required. Using Eq. (423) with $\mathbf{e} = 0.95$, the loss of heat by radiation is

$$q/A = 0.95 \times 1723(0.96^4 - 0.54^4) = 1250 \text{ B.t.u. per hr. per sq. ft.}$$

The total loss of heat from the bare pipe would then be (1250 + 610) or 1860 B.t.u. per hr. per sq. ft. of pipe surface. This calculated result agrees very well with the observed experimental[33] value of 1885 for the same conditions as assumed. In this illustration, it is important to note that the loss of heat by *radiation* makes up about 67 per cent of the total loss, while *free convection* accounts for the remaining 33 per cent. It is evident, then, that radiation is of consequence when the *convection is free.*

3. Now, if instead of still air surrounding the pipe, the air flow be forced at right angles to the pipe with a mass velocity, w/A, of 10 lb. per sec. per sq. ft., with the same instantaneous temperatures as before, the film coefficient of heat transfer by *forced convection* will be found from Eq. (420) to be

$$\mathbf{h} = \frac{0.8 \times 750^{1/3}}{1.315^{0.53}}(10)^{(0.6 + 0.8 \log 1.315)}$$

$$= 31 \text{ B.t.u. per hr. per deg. fahr. per sq. ft.}$$

The rate of heat transfer by *forced convection* for the given temperatures will then be about 13,000 B.t.u. per hr. per sq. ft., while by radiation it will be 1250 B.t.u. per hr. per sq. ft., the same as before. In this instance, the *total rate* of *loss* of heat will amount to 14,250 B.t.u. per hr. per sq. ft., of which *forced convection* is responsible for 91 per cent and radiation for only 9 per cent. Where *convection is forced*, i.e., for high fluid velocities and at temperatures under, say, 800 deg. fahr., radiation usually accounts for a very small portion of the *total* amount of heat transferred.

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[32] See Trans., A.S.M.E., Vol. 48, 1926, p. 921.

[33] See Ind. and Eng. Chem., Vol. 16, 1924, p. 451.

338. Resistance to Heat Transfer.—(a) In engineering apparatus, heat is seldom transferred by conduction, convection, or radiation, each method acting alone, but usually the heat transfer is effected by some combination of these modes.

To indicate the usual complications, consider the very important case of heat transfer from a hot fluid at the temperature, t_1, on one side of a metal plate or tube to another fluid at the lower temperature, t_2, on the other side. This case occurs in the transfer of heat from flue gases to water or steam in boilers and superheaters, and from condensing steam to circulating water in a steam condenser, as well as in other apparatus.

The most satisfactory method of computing the total amount of heat transferred in a unit of time is to obtain an *overall coefficient* of heat transfer, U, and multiply it by the mean temperature difference, θ_m, and by the area A. The overall coefficient is the reciprocal of the overall resistance. *If radiation be ignored*, which is permissible if the temperature is low and the rate of flow high, the evaluation of the overall resistance would be obtained by summing up the individual resistances, in the following manner:

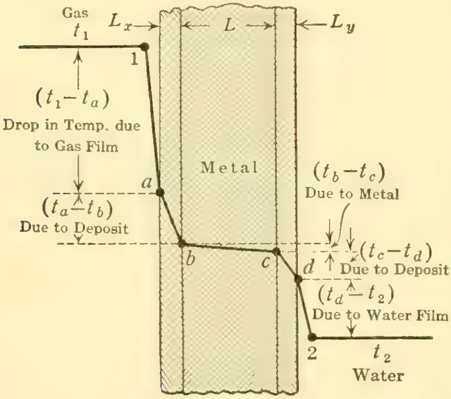

FIG. 492.

The first resistance to heat transfer is between the body of the hot fluid and the surface of contact. This resistance is the *reciprocal of the first film coefficient*, or $1/\mathbf{h}_1$, and is evaluated by the methods given in Sect. **336.** In Fig. 492, this resistance of the fluid film is responsible for the large temperature drop $(t_1 - t_a)$. Next in series comes the resistance offered by any deposit on the metal plate. If the thickness of this deposit be L_x inches, and its specific thermal conductivity be k_x, the first deposit resistance is L_x/k_x. This resistance is responsible for the temperature drop $(t_a - t_b)$, which may or may not be large, depending upon the nature and the thickness of the deposit. Next comes the resistance of the metal plate, which is the ratio of its thickness to its specific thermal conductivity, or L/k. This is responsible for the temperature drop $(t_b - t_c)$, which is usually small, relative to the others. The second deposit offers a resistance, L_y/k_y, causing the temperature drop $(t_c - t_d)$. The second film resistance is the reciprocal of the second film coefficient, or $1/\mathbf{h}_2$, and causes the final temperature drop $(t_d - t_2)$.

If the cross-sectional area of the path perpendicular to the direction of heat flow is constant, as for plane surfaces, the **overall coefficient of heat transfer** for the case analyzed would be

$$U = \frac{1}{1/\mathbf{h}_1 + L_x/k_x + L/k + L_y/k_y + 1/\mathbf{h}_2}. \quad . \quad . \quad (427)$$

The value of U is in B.t.u. per hr. per deg. fahr. per sq. ft. when the film coefficients ($\mathbf{h}_1$ and $\mathbf{h}_2$) are in B.t.u. per hr. per deg. fahr. per sq. ft., the thicknesses (L, L_x and L_y) are in inches and the specific conductivities (k, k_x and k_y) are in B.t.u. per hr. per deg. fahr. per sq. ft. per inch.

(b) The *deposits* of scale, soot, dirt, paint, grease, oil, oxidized metal, or other material on the surface of the metal are usually such that the terms L_x/k_x and L_y/k_y in Eq. (427) are very hard to evaluate, and the film coefficients ($\mathbf{h}_1$ and $\mathbf{h}_2$) must be those determined for *clean* metal plates with no deposits; hence this equation is often difficult to apply. However, if the heat transfer from the body of the fluid to the metal surface is experimentally determined *with the usual deposits*, or if the film coefficient ($\mathbf{h}$) is reduced to allow for fouling, an *overall film coefficient* $\mathbf{h}'$ is obtained and can be used in the following equation:

$$U = \frac{1}{1/\mathbf{h}'_1 + L/k + 1/\mathbf{h}'_2}. \quad . \quad . \quad . \quad . \quad . \quad (428)$$

(c) For thick cylindrical tubes, with fluids inside and outside, the cross-sectional area of the path perpendicular to the flow of heat varies, and the equivalent thickness as given in Sect. **333** (1) must be used. Also, the overall coefficient of heat transfer, U, may be based either upon the *inner* pipe area, in which case it will be called U_1, or upon the *outer* pipe area, in which case it will be called U_2. If the inner pipe radius be r_1 inches, and the outer pipe radius be r_2 inches, then,

$$U_1 = \frac{1}{1/\mathbf{h}_1 + (r_1/k)\log_\epsilon(r_2/r_1) + r_1/r_2\mathbf{h}_2}, \quad . \quad . \quad (429a)$$

and

$$U_2 = \frac{r_1}{r_2}U_1 = \frac{1}{\dfrac{r_2}{r_1\mathbf{h}_1} + \left(\dfrac{r_2}{k}\right)\log_\epsilon(r_2/r_1) + (1/\mathbf{h}_2)}, \quad . \quad (429b)$$

where $\mathbf{h}_1$ = the inner film coefficient of heat transfer, in B.t.u. per hr. per deg. fahr. per sq. ft. of *inner* pipe surface;

$\mathbf{h}_2$ = the outer film coefficient of heat transfer, in B.t.u. per hr. per deg. fahr. per sq. ft. of *outer* pipe surface.

For tubes that are *thin* relative to their diameter, Eq. (428) can be used without serious error.

(d) *Some numerical examples* will now be given to illustrate this method of calculating the overall coefficient of heat transfer.

1. **Ammonia Condenser.**—The tubes are of wrought iron, O.D. = 2.00 in., I.D. = 1.81 in., length = 12.82 ft. The temperature of the entering water is 68 deg. fahr.; the temperature of the condensing ammonia vapor is 82 deg. fahr., and that of the water film may be taken as 77 deg. fahr. The water velocity is 5.6 ft. per second. If the inner tube surface is clean, the water film coefficient is found from Eq. (414) to be

$$\mathbf{h}_1 = \frac{138}{1.81^{0.2}}\left(1 + \frac{50}{12.82/0.1508}\right)\left(\frac{5.6}{0.88}\right)^{0.8}$$

= 853 B.t.u. per hr. per deg. fahr. per sq. ft. of *inner* tube surface.

The ammonia vapor film coefficient, $\mathbf{h}_2$, may be taken as 1635 B.t.u. per hr. per deg. fahr. per sq. ft. of *outer* tube surface. The specific thermal conductivity of wrought iron at 80 deg. fahr. is found from Fig. 479 to be 470 B.t.u. per hr. per deg. fahr. per sq. ft. per in. Then, from Eq. (429*b*), the overall coefficient of heat transfer becomes

$$U_2 = 1 \div \left(\frac{1.104}{853} + \frac{1.00}{470}\log_e 1.104 + \frac{1}{1635}\right)$$

= 470 B.t.u. per hr. per deg. fahr. per sq. ft. of *condensing surface,*

and

$U_1 = 1.104 \times 470 = 520$ B.t.u. per hr. per deg. fahr. per sq. ft. of *inner* tube surface.

2. **Steam Condenser.**—The tubes are of brass, O.D. = 1.00 in., I.D. = 0.902 in., length = 65 in. The temperatures of the entering water and of the condensing steam are 56 and 126 deg. fahr., respectively. The water film temperature may be taken as 84 deg. fahr. The water velocity is 3 ft. per sec. From Sect. **336** (**c**), the value of $\mathbf{h}_1$ is 670 B.t.u. per hr. per deg. fahr. per sq. ft. of *inner* tube surface, if the inner tube surface is clean. If the inner surface is foul, this value must be reduced. For some scale on the outer surface, the value of $\mathbf{h}_2$, the film coefficient for condensing steam may be taken as 2000 B.t.u. per hr. per deg. fahr. per sq. ft. of condensing surface, as previously suggested. At a temperature of 120 deg. fahr., the specific thermal conductivity of 70:30 brass would be 780. From Eq. (429*b*),

$$U_2 = 1 \div \left(\frac{1.108}{670} + \frac{0.5}{780}\log_e 1.108 + \frac{1}{2000}\right)$$

= 450 B.t.u. per hr. per deg. fahr. per sq. ft. of condensing surface,

and

$U_1 = 1.108 \times 450 = 500$ B.t.u. per hr. per deg. fahr. per sq. ft. of inner tube surface.

(**e**) These examples illustrate how the *overall coefficient of heat transfer for any apparatus may be logically derived* when there is sufficient information available to predict the individual resistances to heat transfer. However, information as to the exact value of the film coefficients for all fluids flowing under all conditions is very incomplete. Furthermore, the deposits on the surfaces of metal tubes introduce an additional uncertainty into such calculations. Although such a logical study of the individual resistances shows the factors that enter into the problem, and enables one to design new apparatus, it may be preferable to use the **overall coefficient as obtained from reliable test data** for the design of heat transfer apparatus, *when such equipment is similar to existing types and operates under the same conditions.*

For example, very extensive tests on the overall rate of heat transfer in *boiler tubes* have been conducted by the Babcock and Wilcox Company.[34] The results of the test indicate that the overall coefficient of heat transfer in **boiler tubes** with the gas flow at right angles to the tube may be given by the following empirical equation:

$$U = 2 + 5.04\left(\frac{w}{A}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad (430a)$$

where

U = the overall coefficient of heat transfer, in B.t.u. per hr. per deg. fahr. per sq. ft.;

$\frac{w}{A}$ = the mass velocity of the gas in lbs. per sec. per sq. ft. of net area between the tubes in one row (average of all passes).

This equation is of the general form,

$$U = a + b(v), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (430b)$$

which is in accordance with Reynolds's Theory, as stated in Sect. **334** (**c**).

For the heat transfer in boiler tubes with the same conditions as in the tests, such an empirical equation for the overall coefficient may be more satisfactory for design or for predicting performance than any effort to secure the individual resistances to heat flow.

Granting now that a reasonable value of the overall coefficient of heat transmission may be obtained from test data, or from calculations or by a combination of both, it is next necessary to find the mean temperature difference, θ_m, of the two fluids between which the transmission of heat is taking place. In general, this is not the simple matter that might be expected because the temperature of each fluid may vary, as shown by the five cases in the next section.

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[34] See "Experiments in the Rate of Heat Transfer from a Hot Gas to a Cooler Metallic Surface." Published in 1916 by the B. & W. Co.

339. Heat Transmission in Engineering Apparatus.—(a) In the preceding section, the transfer of heat from a hot fluid at a uniform temperature to a cold fluid at a uniform lower temperature has been considered. In most heat exchange apparatus, however, the temperature of each of the fluids may not be uniform, but may change continuously in the direction of flow.

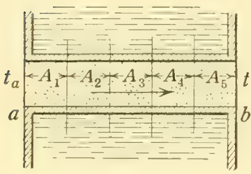

Fig. 493.

Imagine, for instance, that the tube in Fig. 493 is surrounded with boiling water (temperature constant) and that hot gas flows from a to b, becoming cooler as it progresses. Then the average difference of temperature (θ_{m_1}) through the wall of surface A_1 is greater than that (θ_{m_2}) at surface A_2, and the latter is greater than that (θ_{m_3}) at surface A_3, and so on through the length of the tube. With a tube of infinite length the gas could theoretically be cooled to the temperature of the water, and the temperature difference at the end b would be zero. Thus $\theta_{m_1} > \theta_{m_2} > \theta_{m_3} > \ldots \theta_{m_n}$ and each portion of the surface is less effective than those preceding and more so than those following.

Since the curve of temperature change of the gases is not straight, the mean temperature difference for the surface as a whole is not one-half the sum of the initial and final differences. Before the case can be analyzed mathematically, it will be necessary to find the true value of the *mean temperature difference.*

The five cases following need to be considered because they all occur frequently in engineering work, and it is essential to establish a convenient system of notation that may be applied to each case.

(b) **Case I. Hot fluid at constant temperature.**—*A hot fluid at constant temperature surrenders heat to a flowing cold fluid, whose temperature is increased.* Surface condensers and feedwater heaters are examples of this case, for in both of these, heat from the exhaust or bled steam (at constant temperature) is surrendered to water which is raised in temperature as it flows through the apparatus. This case is shown by the curves in Fig. 494, in which ordinates are temperatures and abscissas are extent of surface. The upper curve is for the hot fluid and the lower for the cold one, the flow being toward the right. It will be seen that the final temperature of the cold body depends on the total length of surface, and that, as the flow progresses, the temperature of the cold medium gradually approaches that of the hot fluid, and the temperature difference and value of each unit of surface becomes less.

(c) **Case II. Cold fluid at constant temperature.**—*A fluid at constant temperature receives heat from a hotter flowing fluid whose temperature* *decreases.* An example of this is the steam boiler, in which the boiling water (at constant temperature) receives heat from the hot gases which decrease in temperature as they progress. Fig. 495 is the diagram for this case.

(d) **Case III. Parallel flow.**—The term is understood by engineers to mean parallel flow in the *same* direction on opposite sides of the plate. *The hot and cold fluids both flow in the same direction, their temperatures converging nearer to equality as they progress,* as shown in Fig. 496, in which the arrows show the direction of flow.

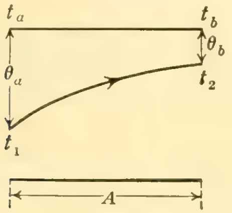

Fig. 494.

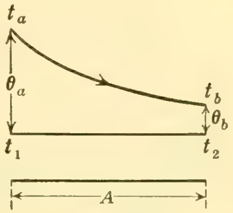

Fig. 495.

(e) **Case IV. Counter flow.**—*The hot and cold fluids flow in opposite directions, the exit temperature of each fluid approaching the entering temperature of the other as they proceed.* This is shown in Fig. 497 in which the directions of flow are shown by the arrows, t_a being the initial temperature of the hot fluid and t_1 being that of the cold one. The relation between the heat absorbing, or surrendering, capacities of the two fluids determines whether the two curves diverge or converge or are parallel.

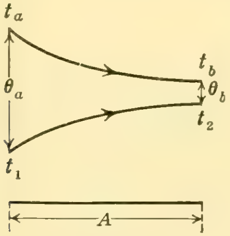

Fig. 496.

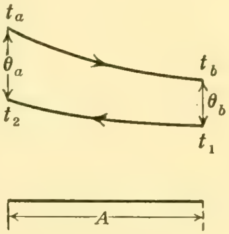

Fig. 497.

(f) **Case V. Constant temperature of both fluids.**—*Transmission of heat from a hot fluid of constant temperature to a cold one which remains at a lower constant temperature.* The vacuum pan is an example of this case, since the latent heat of the steam (at constant temperature) furnishes the heat for evaporating (at constant lower temperature) the other medium.

(g) The **general notation** which will be applied herein to all cases of heat transmission between two flowing fluids is as follows:

t_a = temperature of the *hot fluid entering* the apparatus, in deg. fahr.;
t_b = temperature of the *hot fluid leaving* the apparatus, in deg. fahr.;
t_1 = temperature of the *cold fluid entering* the apparatus, in deg. fahr.;
t_2 = temperature of the *cold fluid leaving* the apparatus, in deg. fahr.;
θ_a = difference in temperature of *entering hot fluid* and the surrounding cold fluid;
$= t_a - t_1$ for parallel flow;
$= t_a - t_2$ for counter flow;
θ_b = difference in temperature of *leaving hot fluid* and the surrounding cold fluid;
$= t_b - t_2$ for parallel flow;
$= t_b - t_1$ for counter flow;
θ_m = mean temperature difference between the two fluids while passing through the apparatus;
c_h = mean specific heat of the hot fluid;
c_c = mean specific heat of the cold fluid;
w_h = weight of hot fluid flowing per unit time, lb. per hr.;
w_c = weight of cold fluid flowing per unit time, lb. per hr.;
U = thermal transmittance, or overall coefficient of heat transmission, in B.t.u. per hr. per deg. fahr. per sq. ft.;
q = quantity of heat transmitted per unit time from the hot fluid to the cold one, in B.t.u. per hr.;
$r_f = w_h c_h / w_c c_c$ = the fluid ratio;
A = total surface in contact with either fluid, in sq. ft.

(h) The **quantity of heat transmitted**, q, per unit time may now be expressed very conveniently by two entirely independent equations, the first one being dependent upon the law of conservation of energy, and the second one derived from the laws of heat transmission.

If the heat transfer apparatus be so arranged that there is no loss of heat externally, it follows that the amount of heat given up by the hot fluid must be just equal to that received by the cold one in the same unit of time or,

$$q = w_h c_h (t_a - t_b) = w_c c_c (t_2 - t_1). \quad . \quad . \quad . \quad . \quad (431)$$

From what has previously been given regarding the overall coefficient of heat transmission, the heating surface involved and the mean temperature difference between the two fluids while passing through the apparatus, it follows that,

$$q = UA\theta_m \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (432)$$

The proper value of U to use under a given set of conditions is sometimes difficult to determine, but the previous discussion shows how it may be evaluated fairly closely by logical means in many cases. The values given in Table XXIV are arranged for convenient reference to show the wide variation in the magnitude of U under the great variety of operating conditions. These coefficients are given further consideration in the several chapters relating to each piece of apparatus.

TABLE XXIV

Overall Coefficients of Heat Transmission in Engineering Apparatus

Apparatus	Chief Means of Heat Transmission	Chief Factors Affecting Overall Coefficient	Overall Coefficient, U, in B.t.u. per Hour per Degree Fahrenheit Mean Temperature Difference per Square Foot of Surface
Steam boiler tubes...	Convection Radiation Conduction	Gas velocity Gas Temperature Deposits on tubes Scale in tubes Water velocity	3–14
Steam superheaters..	Radiation or Convection Conduction	Gas temperature Gas velocity Deposits on tubes Scale in tubes Steam velocity	8–15 when next to furnace. 1–8 otherwise
Economizers........	Convection Conduction	Gas velocity Deposits on tubes Water velocity Scale in tubes	2–10
Surface condensers...	Convection Conduction	Velocity of circulating water Air in steam Deposits in tubes	200–1000
Air preheaters.......	Convection Conduction	Gas velocity Air velocity	1–4
Feedwater heaters...	Convection Conduction	Water velocity Deposits on tubes Air in steam (at low pressures)	200–1500

Regarding the mean temperature difference θ_m, it may be readily observed from Figs. 494 to 497 that the changing temperature of the flowing fluid is not a linear function of surface or distance traveled through the apparatus. The rate of heat transmission depends upon the temperature difference, but the transfer of heat causes the temperature of the two fluids to approach equality. Consequently, as the temperature difference becomes less, the two fluid temperatures converge less rapidly in all cases. It therefore follows that the mean temperature difference, θ_m, cannot be the simple *arithmetic mean* of θ_a and θ_b. However, it can be proved, in a manner similar to that used in the next section, that for all cases *involving a change of temperature of either one or both fluids* under the conditions specified in Sect. **340(b), the true** or so-called **logarithmic mean temperature difference** [35] **is**

$$\theta_m = \frac{\theta_a - \theta_b}{\log_\epsilon \theta_a/\theta_b} = \frac{\theta_a - \theta_b}{\log_\epsilon \theta_a - \log_\epsilon \theta_b}. \quad . \quad . \quad . \quad (433)$$

(i) The **amount of heating surface** required to transmit heat at a certain rate (B.t.u. per hour) is commonly needed by engineers. This surface in square feet is, evidently, from Eqs. (432) and (433),

$$A = \frac{q}{U\theta_m} = \frac{q}{\dfrac{U(\theta_a - \theta_b)}{\log_\epsilon \theta_a - \log_\epsilon \theta_b}} = \frac{q \log_\epsilon (\theta_a/\theta_b)}{U(\theta_a - \theta_b)}. \quad . \quad . \quad (434)$$

(j) A **chart for finding** θ_m is given in Fig. 498. By using the proper one of the three scales given in this chart any range of temperatures that is likely to be encountered may be readily and conveniently handled without any calculation, as shown by the three examples given on the chart.

340. The Logarithmic Mean Temperature Difference.[36]—(a) The greatest discrepancy between the logarithmic and arithmetic mean difference in temperature occurs with parallel flow. This case may therefore very properly be taken as a suitable one for which to derive θ_m in terms of θ_a and θ_b, and thus also show the general method of procedure applicable to the others.

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[35] Under a narrow range of conditions there may be a fairly constant ratio between the arithmetic and the logarithmic mean temperature difference, and in that case it may be more convenient to use the more easily found **arithmetic mean,** but with a modified value of U.

[36] This section may be omitted in short courses, if Eq. (433) be accepted without proof.

CHART TO FIND
LOGARITHMIC MEAN TEMPERATURE DIFFERENCE

$$\theta_m = \frac{\theta_a - \theta_b}{\log_\varepsilon \dfrac{\theta_a}{\theta_b}},$$

Where θ_m = logarithmic mean temp. diff.

θ_a = temp. of entering hot fluid (t_a) — temp. of surrounding cold fluid.

θ_b = temp. of leaving hot fluid (t_b) — temp. of surrounding cold fluid.

Directions: Any straight line cutting the curved axis at values of θ_a and θ_b will intersect straight axis in desired value of θ_m, all being on same scale.

EXAMPLES

With Scale I, $\theta_a = 30$, $\theta_b = 10$, $\theta_m = 18.2$
" " II, $\theta_a = 150$, $\theta_b = 50$, $\theta_m = 91$
" " III, $\theta_a = 1500$, $\theta_b = 500$, $\theta_m = 910$

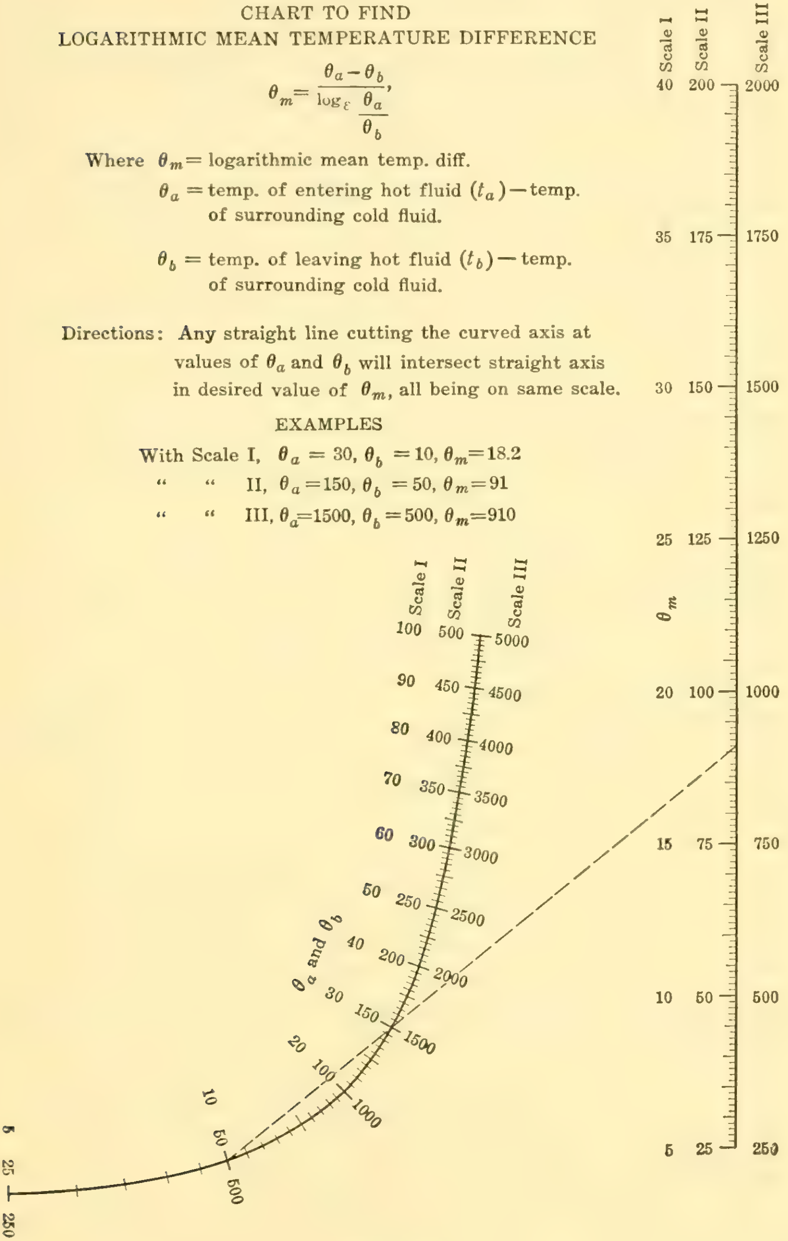

FIG. 498.

(b) *To prove that* $\theta_m = (\theta_a - \theta_b) \div \log_e(\theta_a/\theta_b)$, consider Case III as illustrated again in Fig. 499, and assume (1) that each fluid has constant specific heat over the temperature range in question, and (2) that at all sections of the apparatus there is the same value of U, or the same resistance to heat flow. Then the heat added to the cold fluid to raise its temperature δt_c deg., is,

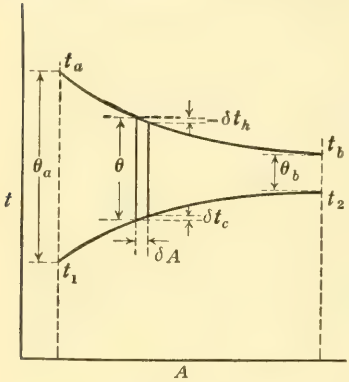

FIG. 499.

$$\delta q_c = w_c c_c \delta t_c. \quad . \quad . \quad . \quad (A)$$

Correspondingly, for the hot fluid the heat abstracted by changing its temperature $-\delta t_h$ degrees is

$$\delta q_h = w_h c_h(-\delta t_h). \quad . \quad . \quad (B)$$

Hence, by Eq. (431)

$$w_c c_c \delta t_c = - w_h c_h \delta t_h, \quad \text{or} \quad \frac{\delta t_h}{\delta t_c} = - \frac{w_c c_c}{w_h c_h}.$$

Subtracting unity from each side,

$$\frac{\delta t_h - \delta t_c}{\delta t_c} = \frac{-(w_c c_c + w_h c_h)}{w_h c_h}.$$

But from Fig. 499, $\delta\theta = \delta t_h - \delta t_c$. Substituting this and transposing gives,

$$\delta t_c = -\left(\frac{w_h c_h}{w_c c_c + w_h c_h}\right)\delta\theta. \quad . \quad . \quad . \quad . \quad . \quad . \quad (C)$$

Substituting in Eq. (A), and changing to the differential form,

$$dq_c = -\left[\frac{w_c c_c w_h c_h}{w_c c_c + w_h c_h}\right] d\theta = - K d\theta, \quad . \quad . \quad (D)$$

in which K is the bracketed quantity in the first part of the equation. Therefore,

$$q_c = -K\int_{\theta_a}^{\theta_b} d\theta = K(\theta_a - \theta_b). \quad . \quad . \quad . \quad . \quad (E)$$

From Eq. (432) $q = UA\theta_m$; therefore

$$\theta_m = K\frac{(\theta_a - \theta_b)}{UA}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (F)$$

But it is still desirable to determine θ_m in terms of θ_a and θ_b alone.

From Eq. (D) and $dq = U\theta \cdot dA,\ U\theta \cdot dA = -K \cdot d\theta.$

Therefore

$$U\int_0^A dA = -K\int_{\theta_a}^{\theta_b}\frac{d\theta}{\theta} = K\int_{\theta_b}^{\theta_a}\frac{d\theta}{\theta},$$

or

$$UA = K \cdot \log_\epsilon \frac{\theta_a}{\theta_b}. \quad \text{. . . . .} \quad (G)$$

Substituting this value in Eq. (F) gives

$$\theta_m = \frac{\theta_a - \theta_b}{\log_\epsilon (\theta_a/\theta_b)},$$

which was given as Eq. (433) and which it was desired to prove.

Although Eq. (433) has been derived only for Case III, it can be shown that this same expression for θ_m holds also for the other cases involving a change in temperature with c_h, c_c, and U constant.[37]

341. Effectiveness or Performance of Heat-Transmitting Surfaces.—**(a)** It has been seen that the rate of transmission of heat through a plate depends directly on the difference between the temperatures of its two surfaces. When the temperatures of the fluids on each side of the plate are maintained constant, the temperature drop is the same at all points over the surface, hence the rate of transmission and effectiveness of surfaces is uniform over the whole area. But when there is flow of one or both of the fluids with a change in temperature, as is generally the case, the conditions are quite different. Thus the amount of heat transmitted increases as the surface is enlarged but not in direct proportion, because the rate of transmission through each additional increment of area is decreased as the temperatures of the two fluids converge. The cost of heating surface, auxiliary equipment, attendance, maintenance, etc., increases almost in direct proportion to the amount of surface. Consequently, a certain extent of surface in all cases will be most economical, because if this amount be exceeded, the value of the additional heat transmitted will not pay for the additional surface installed. It is always important to engineers, therefore, to express the effectiveness, or performance, of heating surfaces in such a way that a just comparison may be made of different installations. Sometimes the comparison can best be made on the basis of the overall coefficient of heat transmission, or again it may be preferable to calculate the efficiency of the heating surface. Both of these methods will now be discussed.

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[37] For proof covering Cases I, II, and IV, see "Heat Transmission by Radiation, Conduction, and Convection," Royds, pp. 143 and 144.

For those who use this reference special attention is called to the fact that with Royds' notation, t_1 is the temperature of the cold fluid *entering the apparatus for parallel flow and leaving the apparatus for counter flow*, whereas in this text, t_1 represents the temperature of the cold fluid *entering the apparatus* in all cases, as given in Sect. **339** **(g)**.

(b) **The overall coefficient** of heat transmission, U, which may be secured from the test of given apparatus, is a very satisfactory measure of its performance. For example, a new surface condenser may be tested with measured rates of steam and water flow and with the temperatures of the entering steam and water both known quantities. Then, by measuring the exit temperatures of the two fluids, the overall coefficient of heat transmission, U, expressed in B.t.u. per hr. per deg. fahr. mean temperature difference, per sq. ft., may be calculated. After the apparatus has been in service some time, another test may be made with approximately the same rates of steam and water flow, the same entering temperature as before, and the new overall coefficient of heat transmission may be secured. A comparison of these coefficients is then a direct measure of the performance of the condenser in the two conditions, and the effects of deposits of scale, dirt, oil, etc., on the tubes, and of occluded air in the steam, are clearly shown by a lowering in the value of U.

The overall-coefficient method is basically the only one available for the comparison of equipment in which neither fluid changes its temperature, as in *Case V*. But for such equipment, instead of determining the value of U, the results of tests may be expressed on the basis of the difference in temperature of the two fluids required to give the same rate of heat transmission; the method is fundamentally the same.

On the other hand, if either fluid changes its temperature in passing through the heat transfer apparatus, the performance may then be expressed either by the coefficient of transmission, or by the efficiency as defined in the next paragraph.

(c) **The definition of efficiency of a heat-transmitting surface** is not so simple as might at first be supposed. It should be remembered that apparatus of this character is used, in one instance, *to deliver heat energy to the cold fluid*, and in another, *to abstract heat energy from the hot fluid*. The steam boiler and condenser afford typical illustrations of these respective uses. For all cases, the heat added to one fluid can equal that abstracted from the other in the apparatus only when there is no transmission of heat to, or from, external bodies. In some instances, such as boilers for example, this loss may be appreciable. Even though the waste of heat to external bodies be considered zero, the ideal apparatus may not have an efficiency of 100 per cent, because a different arrangement of the same extent of heating surface might give still better results.

Thus it is necessary *to base the efficiency on the amount of heat energy available for transfer under the given conditions.* This amount is determined by the product of *rate of flow of one* of the fluids, *its specific heat,* and the *difference in temperature* of the hot and cold fluids at entrance, or $(t_a - t_1)$. This particular difference in temperatures should be used because t_a and t_1 represent the limits that would be approached by the cold and hot fluids, respectively, if they were passed over an infinite extent of heating surface arranged in the best way possible, if there were no loss of heat to external bodies, and if the heat-absorbing capacities of these two fluids were the same.

If for each degree change of temperature, the heat-absorbing capacity of the cold fluid $(w_c c_c)$ is less than that of the hot fluid $(w_h c_h)$, the exit temperature of the hot fluid (t_b) cannot be brought down to that of the entering cold fluid (t_1). Similarly, if the heat-absorbing capacity of the cold fluid is greater than that of the hot fluid the exit temperature of the cold fluid (t_2) cannot be brought up to that of the entering hot fluid (t_a).

In many engineering applications the heat-absorbing capacities of the two fluids are not equal, and consequently in such cases $(t_a - t_1)$ must be multiplied either by $w_c c_c$, or $w_h c_h$, whichever is the smaller, to give the amount of heat available for transfer in any particular case.

Therefore, the efficiency of the heating surfaces in Cases I, II, III, and IV may be expressed as,

$$e = \frac{\text{Quantity of heat usefully transmitted to or from one fluid in unit time.}}{\text{Amount available in unit time, with given temp. limits and rate of fluid flow.}}$$

If $w_h c_h < w_c c_c$, and if q_u represents the numerator,

$$e = \frac{q_u}{w_h c_h (t_a - t_1)}, \qquad \text{(435a)}$$

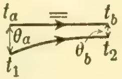
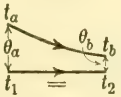
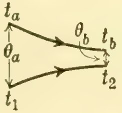
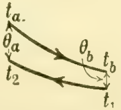
or, if $w_c c_c < w_h c_h$,

$$e = \frac{q_u}{w_c c_c (t_a - t_1)}. \qquad \text{(435b)}$$

When these equations are applied to cases such as I and II, in which the hot or cold fluid, respectively, remains at constant temperature, the corresponding value of the specific heat of this fluid is infinite; and consequently the other fluid determines the amount of heat that is available for transmission. Note that the rate of flow of the fluid that remains at constant temperature must be sufficient to maintain constant pressure on its side of the heat-transfer surface, otherwise its temperature could not be kept constant.

TABLE XXV *

EFFICIENCIES OF HEAT TRANSMISSION SURFACES

Purpose	Neglecting External Losses	Including External Losses †	
	To Transmit Heat to or from Either Fluid	To Add Heat to the Cold Fluid	To Abstract Heat from the Hot Fluid
Case I  t_a = t_b θ_a θ_b t_2 t_1	$e = \frac{t_2 - t_1}{t_a - t_1} = \frac{\theta_a - \theta_b}{\theta_a}$, since $t_1 = t_b$	$e = \frac{t_2 - t_1}{t_a - t_1} = \frac{\theta_a - \theta_b}{\theta_a}$	$e = \frac{t_2 - t_1}{t_a - t_1} - \frac{L'}{w_c c_c (t_a - t_1)}$
Case II  t_a θ_a θ_b t_b t_1 = t_2	$e = \frac{t_a - t_b}{t_a - t_1} = \frac{\theta_a - \theta_b}{\theta_a}$, since $t_1 = t_2$	$e = \frac{t_a - t_b}{t_a - t_1} - \frac{L}{w_h c_h (t_a - t_1)}$	$e = \frac{t_a - t_b}{t_a - t_1} = \frac{\theta_a - \theta_b}{\theta_a}$
Case III  t_a θ_a θ_b t_b t_2 t_1	$e = \frac{t_2 - t_1}{t_a - t_1}$ if $w_c c_c < w_h c_h$ $e = \frac{t_a - t_b}{t_a - t_1}$ if $w_h c_h < w_c c_c$	$e = \frac{t_2 - t_1}{t_a - t_1}$ if $w_c c_c < w_h c_h$ $e = \frac{t_a - t_b}{t_a - t_1} - \frac{L}{w_h c_h (t_a - t_1)}$ if $w_h c_h < w_c c_c$	$e = \frac{t_2 - t_1}{t_a - t_1} - \frac{L'}{w_c c_c (t_a - t_1)}$ if $w_c c_c < w_h c_h$ $e = \frac{t_a - t_b}{t_a - t_1}$ if $w_h c_h < w_c c_c$
Case IV  t_a θ_a θ_b t_b t_2 t_1	$e = \frac{t_2 - t_1}{t_a - t_1}$ if $w_c c_c < w_h c_h$ $e = \frac{t_a - t_b}{t_a - t_1}$ if $w_h h_h < w_c c_c$	$e = \frac{t_2 - t_1}{t_a - t_1}$ if $w_c c_c < w_h c_h$ $e = \frac{t_a - t_b}{t_a - t_1} - \frac{L}{w_h c_h (t_a - t_1)}$ if $w_h c_h < w_c c_c$	$e = \frac{t_2 - t_1}{t_a - t_1} - \frac{L'}{w_c c_c (t_a - t_1)}$ if $w_c c_c < w_h c_h$ $e = \frac{t_a - t_b}{t_a - t_1}$ if $w_h c_h < w_c c_c$

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* This table includes all cases involving a change of temperature of either fluid.

† L represents the absorption of heat by the atmosphere from the hot fluid, in B.t.u. per unit time. L' represents the absorption of heat by the cold fluid from the atmosphere in B.t.u. per unit time; w_h and w_c represent the weights of hot and cold fluids flowing per unit of time.

(d) **A summary of efficiencies** of heat-transmitting surfaces for the different cases, as determined by Eq. (435), is shown in Table XXV. It should be noted that in all cases the efficiency is given in terms of the temperatures attained by the two fluids, which in turn depend mainly upon the extent and condition of the surface and upon the absorbing capacities and velocities of the fluids.

Cases I and II, in which one fluid undergoes no change in temperature, are of special interest in dealing with the question of how to determine the amount of heat available for transfer. For example, consider Case I, in which it is desired to add heat *to a cold fluid.* Then, by the method recommended, and as given in Table XXV,

$$e = \frac{w_c c_c (t_2 - t_1)}{w_c c_c (t_a - t_1)} = \frac{t_2 - t_1}{t_a - t_1} = \frac{\theta_a - \theta_b}{\theta_a}, \quad . \quad . \quad . \quad (436)$$

which is an expression that approaches a value of 100 per cent when θ_b approaches zero, as is possible with an infinite extent of surface and a sufficient supply of vapor maintained at $t_a (= t_b)$ even though there may be a loss to external bodies.

For *Case III*, parallel flow, the efficiency, as given in Table XXV, can never become 100 per cent even though there be no loss to external bodies and an infinite extent of surface be used. The best that can be done with this arrangement of heating surface is to have the two fluids approach a common temperature, say t_x, with the corresponding value of maximum efficiency of $\frac{t_x - t_1}{t_a - t_1}$, if $w_c c_c < w_h c_h$, or $\frac{t_a - t_x}{t_a - t_1}$ if $w_h c_h < w_c c_c$.

For *Case IV*, counter flow, the two fluids will have a constant difference in temperature maintained between them, or $\theta_a = \theta_b$, provided their heat-absorbing capacities are equal and there are no losses. If $w_h c_h$ is larger than $w_c c_c$, θ_a is less than θ_b, which means that the two curves converge at the high temperature end. If an infinite extent of surface be used to add heat to the cold fluid, it may be heated to t_a and then the efficiency would be 100 per cent. If $w_c c_c$ is larger than $w_h c_h$ the curves converge at the low temperature end and therefore t_b approaches t_1 as the surface approaches infinity, if the loss is zero.

Table XXV also gives efficiency equations which apply when heat is lost to the atmosphere by the hot fluid and when it is absorbed from the atmosphere by the cold fluid.

342. Surfaces Required for Counter Flow and Parallel Flow.—(a) *The amount of heating surface required to transmit a given amount of heat per unit of time* with counter flow, as compared with that required with parallel flow, is often of importance in engineering applications.

To make a fair comparison, it is essential that the two classes of heating surface shall operate under *similar conditions*, which means that the following values must be the same for each heating surface: (1) The entering temperature of the hot fluid (t_a). (2) The entering temperature of the cold fluid (t_1). (3) The composition of the fluids and the weight of flow per unit time. (4) The composition, thickness and degree of cleanliness of the heating surfaces. (5) The velocity of the hot fluid. (6) The velocity of the cold fluid.

(b) Regardless of whether there are any losses involved, it is stipulated that the same amount of heat is to be transmitted from the hot fluid to the cold one; consequently from conditions (1), (2) and (3) above and Eq. (431), it follows that the same final temperature of the cold fluid (t_2) is to be attained with both arrangements of heating surface. Whether the same final temperature of the hot fluid (t_b) is reached with each heating surface depends upon the external losses, which are usually so small that they may be ignored in this comparison, and then the exit temperature of the hot fluid (t_b) will be the same for the two surfaces. Let the subscripts c and p be used in this discussion to designate counter flow and parallel flow, respectively. Then from Eq. (432),

$$q = U_p A_p \theta_{m_p} = U_c A_c \theta_{m_c} \quad . \quad . \quad . \quad . \quad . \quad . \quad (437)$$

Since the overall coefficient of heat transmission, U, depends chiefly upon the nature of the fluids, their velocities, and the contact surface, it follows that for this comparison U_p and U_c may be considered equal. Therefore,

$$A_p \theta_{m_p} = A_c \theta_{m_c},$$

or,

$$\frac{A_p}{A_c} = \frac{\theta_{m_c}}{\theta_{m_p}}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (438)$$

which states mathematically that, external losses being negligible, to transmit the same quantity of heat per unit time under similar conditions, the surface required with parallel flow is to the surface required with counter flow inversely as their logarithmic mean temperature differences. Expanding Eq. (438) gives

$$\frac{A_p}{A_c} = \frac{[(t_a - t_2) - (t_b - t_1)] \div \log_\epsilon [(t_a - t_2)/(t_b - t_1)]}{[(t_a - t_1) - (t_b - t_2)] \div \log_\epsilon [(t_a - t_1)/(t_b - t_2)]}. \quad . \quad (439)$$

This equation determines the ratio (A_p/A_c) of the surface required with parallel flow to that required with counter flow for similar conditions, in terms of the entering and exiting temperatures of both fluids. The important applications are those in which both fluids are changing temperature and then the surface required with parallel flow will always be larger than that needed for counter flow, as is proved in the next section by showing that for such cases $\theta_{m_c} > \theta_{m_p}$. If either fluid remains at constant temperature, Eqs. (438) and (439) will reduce to unity since the direction of flow of one fluid relative to the other can not then affect the mean temperature difference or the surface required.

(c) The general proof that $(\theta_{m_c}/\theta_{m_p}) > 1$ for similar conditions involving a change of temperature of both fluids follows:

$$\frac{1}{\theta_{m_p}} = \frac{\log_\epsilon\left(\frac{t_a - t_1}{t_b - t_2}\right)}{(t_a - t_1) - (t_b - t_2)}; \quad \text{and} \quad \frac{1}{\theta_{m_c}} = \frac{\log_\epsilon\left(\frac{t_a - t_2}{t_b - t_1}\right)}{(t_a - t_2) - (t_b - t_1)}.$$

Let $x = \frac{t_a - t_1}{t_b - t_2}$, then $\frac{x - 1}{x + 1} = \frac{t_a - t_b - t_1 + t_2}{t_a + t_b - t_1 - t_2}$.

$$\text{Since } \log_\epsilon x = 2\left[\left(\frac{x-1}{x+1}\right) + \frac{1}{3}\left(\frac{x-1}{x+1}\right)^3 + \frac{1}{5}\left(\frac{x-1}{x+1}\right)^5 + \text{etc.}\right],$$

$$\frac{1}{\theta_{m_p}} = \frac{\log_\epsilon x}{t_a - t_b - t_1 + t_2}$$

$$= \frac{1}{t_a + t_b - t_1 - t_2} \cdot \frac{2}{\left(\frac{t_a - t_b - t_1 + t_2}{t_a + t_b - t_1 - t_2}\right)} \left[\left(\frac{x-1}{x+1}\right) + \frac{1}{3}\left(\frac{x-1}{x+1}\right)^3 + \frac{1}{5}\left(\frac{x-1}{x+1}\right)^5 + \text{etc.}\right]$$

$$= \frac{2}{t_a + t_b - t_1 - t_2}\left[1 + \frac{1}{3}\left(\frac{x-1}{x+1}\right)^2 + \frac{1}{5}\left(\frac{x-1}{x+1}\right)^4 + \text{etc.}\right].$$

Let $y = \frac{t_a - t_2}{t_b - t_1}$, then $\frac{y - 1}{y + 1} = \frac{t_a - t_b + t_1 - t_2}{t_a + t_b - t_1 - t_2}$.

$$\text{Since } \log_\epsilon y = 2\left[\left(\frac{y-1}{y+1}\right) + \frac{1}{3}\left(\frac{y-1}{y+1}\right)^3 + \frac{1}{5}\left(\frac{y-1}{y+1}\right)^5 + \text{etc.}\right],$$

$$\frac{1}{\theta_{m_c}} = \frac{\log_\epsilon y}{t_a - t_b + t_1 - t_2}$$

$$= \frac{1}{t_a + t_b - t_1 - t_2} \cdot \frac{2}{\left(\frac{t_a - t_b + t_1 - t_2}{t_a + t_b - t_1 - t_2}\right)} \left[\left(\frac{y-1}{y+1}\right) + \frac{1}{3}\left(\frac{y-1}{y+1}\right)^3 + \frac{1}{5}\left(\frac{y-1}{y+1}\right)^5 + \text{etc.}\right]$$

$$= \frac{2}{t_a + t_b - t_1 - t_2}\left[1 + \frac{1}{3}\left(\frac{y-1}{y+1}\right)^2 + \frac{1}{5}\left(\frac{y-1}{y+1}\right)^4 + \text{etc.}\right].$$

Comparing $\frac{1}{\theta_{m_p}}$ and $\frac{1}{\theta_{m_c}}$, $y + 1 = x + 1$, but $(x - 1) > (y - 1)$, since $t_2 > t_1$.

Therefore, $\frac{1}{\theta_{m_p}} > \frac{1}{\theta_{m_c}}$ or $\frac{\theta_{m_c}}{\theta_{m_p}} > 1$. Q.E.D.

(d) Probably the best way to appreciate the significance of Eq. (439) is to prepare a set of curves for some particular values of t_a and t_1, the entering temperatures of the hot and cold fluids, respectively, with a sufficient range of the other variables to cover the field in which one may be interested. An illustration of this is afforded by the curves given in Fig. 500, in which $t_a = 1000$ and $t_1 = 400$ deg. fahr. The ratio A_p/A_c is plotted against the drop in temperature of the hot fluid $(t_a - t_b)$ for two cases. The first set of curves is for different values of the constant temperature rise of the cold fluid $(t_2 - t_1)$, as shown by the solid curves; and the second set is for the constant values of the fluid ratio, r_f, as shown by the broken curves. In applying these curves the simultaneous use of any two of these four variables will determine the value of the other two for the given values of t_a and t_1. For example, assuming $t_a = 1000$ and $t_1 = 400$, then for $r_f = 1$ and $t_a - t_b = 200$, a point is thus located on the chart of Fig. 500, from which one may read from the scale of ordinates that $A_p/A_c = 1.1$, and from the solid curve passing through the same point that $t_2 - t_1 = 200$. If r_f be only 0.5 and the other conditions the same, then $t_2 - t_1 = 100$ and $A_p/A_c = 1.04$. If $r_f = 2$, and it is desired to have the cold fluid increase its temperature 350 deg. $(= t_2 - t_1)$, Fig. 500 shows by interpolation that the drop in temperature of the hot fluid would be 175 deg. and the parallel arrangement of heating surface would require 30 per cent more surface than counter flow arrangement.

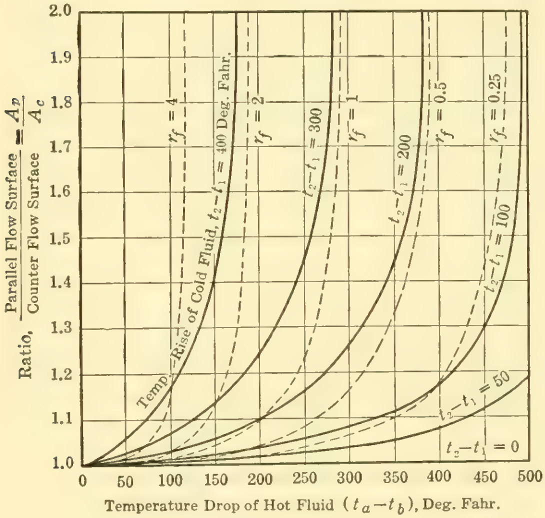

FIG. 500.—Comparison of Surfaces Required to Transmit the Same Amount of Heat with $t_a = 1000$ and $t_1 = 400$ Deg. Fahr.

(e) More generally, Fig. 500 shows that for the given conditions, A_p is always greater than A_c if the temperatures of both fluids change in passing over the heating surface. It also shows that if the temperature of either fluid remains constant (i.e., if either $t_a - t_b = 0$, or $t_2 - t_1 = 0$, as in Cases I and II), it makes no difference as to the surface required to transmit a given quantity of heat, whether the other fluid flows parallel to or counter to the first. This holds true, also, irrespective of the temperature of either fluid.

As the temperature drop of the hot fluid is increased (which may be accomplished by decreasing r_f) the solid lines representing a constant rise in temperature of the cold fluid, $t_2 - t_1$, show that the ratio A_p to A_c increases very rapidly. This ratio will be found to approach as a limit an infinite value in each case when $t_b = t_2$.

In general, when $t_b = t_2$ it follows that $(t_a - t_b) + (t_2 - t_1) = t_a - t_1$; but in Fig. 500, $t_a - t_1 = 1000 - 400 = 600$. Then for this figure, as the ratio A_p/A_c approaches infinity the curve for $t_2 - t_1 = 100$ approaches that for $t_a - t_b = 500$; the curve for $t_2 - t_1 = 200$ approaches that for $t_a - t_b = 400$; and so on for similar combinations.

If r_f is fixed, then as $t_a - t_b$ increases, $t_2 - t_1$ will also increase, and the dotted lines show how marked the advantage of counter flow over parallel flow becomes. As before, when $t_b = t_2$, the ratio of A_p to A_c becomes infinitely large. Obviously, the reason for this is that with parallel flow t_b may be brought down equal to t_2 only with an infinite extent of surface, whereas, with counter flow of the two fluids, t_b may be reduced not only to t_2 but even below t_2 with a finite extent of surface.

A chart similar to Fig. 500 may be readily constructed for any set of conditions as needed.

343. Determination of Factors Giving Desired Efficiency.—(a) It is often desired to have expressions to determine the values of θ_m, A, t_b, and t_2 that will give a certain efficiency, e, with given values of t_1, t_a, U, $w_h c_h$, or $w_c c_c$; also expressions are sometimes needed for determining for any point along the path of the fluid the temperature difference (θ) and the efficiency (e) in terms of extent of surface (A) traversed and in terms of the other quantities initially known. Such expressions will now be derived on the assumption that there is no external transfer of heat to or from either of the fluids, that the overall coefficient of heat transfer (U) is constant throughout the apparatus, and that the specific heats of both fluids are constant. As Cases I and II may be considered as special applications of parallel and counter flow, the expressions for only the latter two need be given.

(b) But before deriving the desired equations it is first necessary to set down the following preliminary relations:

The maximum amount of heat that can be abstracted from the hot fluid by the cold one is

$$q_h = w_h c_h (t_a - t_1). \qquad (440)$$

The maximum amount of heat that the cold fluid can absorb is

$$q_c = w_c c_c (t_a - t_1). \qquad (441)$$

The amount of heat surrendered by the hot fluid, or received by the cold one, is

$$q = w_h c_h (t_a - t_b) = w_c c_c (t_2 - t_1). \qquad (442)$$

Also, the obvious relations given in Part A of Table XXVI for the different values of the fluid ratio $r_f = (w_h c_h)/(w_c c_c)$, will be needed. In this table the equations are designated by numbers (443) to (451), inclusive.

(**c**) Using the foregoing equations and the general one for the mean temperature difference, as given by Eq. (433), the expressions for the area and other factors giving a desired value of the efficiency of surface, and also those for the temperatures of both fluids, the temperature difference and the efficiency resulting after any extent of surface has been traversed, will now be given for one illustrative case, the steps in their derivation from the preceding equations being obvious:

For **parallel flow and $r_f < 1$:** From Eq. (444), $t_b = t_a - e\theta_a$ and from Eq. (445), $t_2 = t_1 + r_f e \theta_a$. Then

$$\theta_m = \frac{\theta_a - (t_b - t_2)}{\log_\epsilon \dfrac{\theta_a}{t_b - t_2}} = \frac{e\theta_a(1 + r_f)}{\log_\epsilon \dfrac{1}{1 - e(1 + r_f)}}, \qquad (a)$$

$$= e\theta_a(1 + r_f) \div \log_\epsilon X_p, \qquad (a')$$

in which

$$X_p = 1 \div [1 - e(1 + r_f)]. \qquad (b)$$

$$A = q/U\theta_m = (e \cdot w_h c_h \theta_a) \div (U\theta_m), \qquad (c)$$

$$= \frac{w_h c_h}{U(1 + r_f)} \cdot \log_\epsilon X_p. \qquad (c')$$

$$e = \left[1 - \epsilon^{-\frac{AU(1 + r_f)}{w_h c_h}}\right] \div (1 + r_f) = (t_a - t_b)/\theta_a. \qquad (d)$$

$$= (1 - \epsilon^{n}) \div (1 + r_f), \qquad (d')$$

in which

$$n = -\frac{AU(1 + r_f)}{w_h c_h}. \qquad (\epsilon)$$

$$t_b = t_a - e\theta_a = t_a - \theta_a[(1 - \epsilon^n) \div (1 + r_f)]. \qquad (f)$$

$$t_2 = t_1 + r_f(t_a - t_b). \qquad (g)$$

From Eqs. (e), (f) and (g), $\theta_b = \theta_a\epsilon^n$; or, in general, the temperature difference, θ, after the fluids have traversed any extent of surface, A, is

$$\theta = \theta_a\epsilon^n. \qquad (h)$$

Before using the forgoing equations it is first desirable to compute the values of X_p and n. These equations are given in Table XXVI as Eqs. (452) to (459), inclusive.

For parallel flow with $r_f = 1$ and $r_f > 1$, and for counter flow with the three different values of r_f, the corresponding equations can be found in a similar manner.

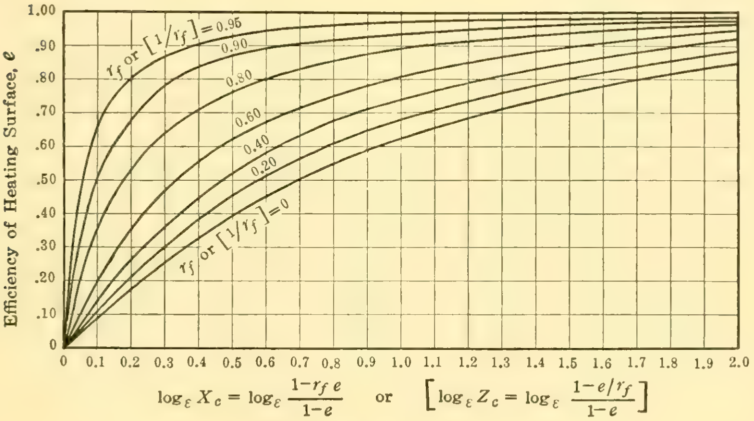

FIG. 501.

(d) For **Cases III and IV,** parallel and counter flow, respectively, all the equations thus derived are given in Table XXVI.

Figure 501 is included to simplify the determination of the values of $\log_\epsilon X_c$ and $\log_\epsilon Z_c$ for any values of e and r_f or $1/r_f$. The bracketed quantities on the curves are used to find $\log_\epsilon Z_c$ and the other quantities to get $\log_\epsilon X_c$. For example, if $r_f = 0.4$ and $e = 0.6$, then $\log_\epsilon X_c = 0.65$; and if $1/r_f = 1/2.5 = 0.4$, and $e = 0.6$, then $\log_\epsilon Z_c = 0.65$.

Note that since A is directly proportional to the abscissa in Fig. 501,

TABLE XXVI

FORMULAS FOR HEAT TRANSMISSION WITH PARALLEL OR COUNTER FLOW

Term Desired	$r_f < 1$	Equation No.	$r_f = 1$	Equation No.	$r_f > 1$	Equation No.
A. Either Parallel or Counter Flow (Case III or IV)						
e	$\frac{q}{q_h} = \frac{t_a - t_b}{t_a - t_1}$	443	$\frac{q}{q_h} = \frac{q}{q_c} = \frac{t_2 - t_1}{t_a - t_1} = \frac{t_a - t_b}{t_a - t_1}$	446	$\frac{q}{q_c} = \frac{t_2 - t_1}{t_a - t_1}$	449
t_b	$t_a - e(t_a - t_1)$	444	$t_a - e(t_a - t_1)$	447	$t_a - \frac{e}{r_f}(t_a - t_1)$	450
t_2	$t_1 + r_f e(t_a - t_1)$	445	$t_1 + e(t_a - t_1)$	448	$t_1 + e(t_a - t_1)$	451
B. Parallel Flow (Case III)						
X_p, Y_p, Z_p	$X_p = 1 \div [1 - e(1 + r_f)]$	452	$Y_p = 1 \div (1 - 2e)$	460	$Z_p = 1 \div [1 - e(1 + 1/r_f)]$	468
n	$-\frac{AU(1 + r_f)}{w_h c_h}$	453	$-\frac{2AU}{wc}$	461	$-\frac{AU(1 + 1/r_f)}{w_c c_c}$	469
θ_m	$e\,\theta_a(1 + r_f) \div \log_e X_p$	454	$2e\,\theta_a \div \log_e Y_p$	462	$e\,\theta_a(1 + 1/r_f) \div \log_e Z_p$	470
A	$\frac{w_h c_h}{U(1 + r_f)} \log_e X_p$	455	$\frac{wc}{2U} \log_e Y_p$	463	$\frac{w_c c_c}{U(1 + 1/r_f)} \log_e Z_p$	471
e	$(1 - \epsilon^n) \div (1 + r_f)$	456	$(1 - \epsilon^n) \div 2$	464	$(1 - \epsilon^n) \div (1 + 1/r_f)$	472

t_b	$t_a - \theta_a[(1 - \epsilon^n) \div (1 + r_f)]$	457	$t_a - \theta_a[(1 - \epsilon^n) \div 2]$	465	$t_a - \frac{t_2 - t_1}{r_f}$	473
t_2	$t_1 + r_f(t_a - t_b)$	458	$t_1 + (t_a - t_b)$	466	$t_1 + \theta_a[(1 - \epsilon^n) \div (1 + 1/r_f)]$	474
θ	$\theta_a \epsilon^n$	459	$\theta_a \epsilon^n$	467	$\theta_a \epsilon^n$	475

C. Counter Flow (Case IV)

X_c, Z_c	$X_c = (1 - r_f e) \div (1 - e)$	476	....	......	$Z_c = (1 - e/r_f) \div (1 - e)$	490
n	$\frac{AU(1 - r_f)}{w_h c_h}$	477	....	......	$\frac{AU(1 - 1/r_f)}{w_c c_c}$	491
θ_m	$e(t_a - t_1)(1 - r_f) \div \log_\epsilon X_c$*	478	$(t_a - t_1)(1 - e)$	484	$e(t_a - t_1)(1 - 1/r_f) \div \log_\epsilon Z_c$*	492
A	$\frac{w_h c_h}{U(1 - r_f)} \log_\epsilon X_c$*	479	$\frac{wc}{U}[e \div (1 - e)]$	485	$\frac{w_c c_c}{U(1 - 1/r_f)} \log_\epsilon Z_c$*	493
e	$(1 - \epsilon^n) \div (r_f - \epsilon^n)$	480	$\frac{AU}{wc} \div \left(1 + \frac{AU}{wc}\right)$	486	$(\epsilon^n - 1) \div (\epsilon^n - 1/r_f)$	494
t_b	$t_a - [(t_a - t_1)(1 - \epsilon^n) \div (r_f - \epsilon^n)]$	481	$t_a - \left[\frac{AU}{wc}(t_a - t_1) \div \left(1 + \frac{AU}{wc}\right)\right]$	487	$t_a - \frac{t_2 - t_1}{r_f}$	495
t_2	$t_1 + r_f(t_a - t_b)$	482	$t_1 + (t_a - t_b)$	488	$t_1 + (t_a - t_1)[(\epsilon^n - 1) \div (\epsilon^n - 1/r_f)]$	496
θ	$(t_a - t_1)[(r_f \epsilon^n - 1) \div (r_f - \epsilon^n)]$	483	$(t_a - t_1)\left(\frac{wc - AU}{wc + AU}\right)$	489	$(t_a - t_1)[(r_f - \epsilon^n) \div (r_f \epsilon^n - 1)]$	497

* Log$_\epsilon$ X_c and log$_\epsilon$ Z_c can be obtained from Fig. 501. (For Cases I and II, see explanation on page 306.)

the curves in this figure show the characteristic relation between efficiency and area of heating surface for different fluid ratios.

The significance of the curve for $1/r_f = 0$ is that the hot fluid is then one that is being condensed at constant temperature, as in Case I; or if $r_f = 0$, the cold fluid is being vaporized at constant temperature, as in Case II.

(e) For **Cases I and II** the equations in parts A and B of Table XXVI also apply, if the following changes are made:

For Case I, make the temperatures t_a and t_b for the hot fluid equal, and use $1/r_f = 0$.

For Case II make the temperatures t_1 and t_2 for the cold fluid equal, and use $r_f = 0$.

The equations for Cases I and II are more easily derived by using the equations for *parallel* flow than by using those given for counter flow. However, it should be noted that if the temperature of either fluid remains constant, the relative directions of fluid flow do not affect the effectiveness of the surface.

(f) 1. To illustrate the use of the equations previously developed, consider a counter flow economizer with $w_h = 96{,}000$ lb. per hr., $c_h = 0.25$, $w_c = 64{,}000$ lb. per hr., $c_c = 1.0$, $t_a = 900$ deg. fahr., $t_1 = 100$, and $U = 8$. Find the extent of heating surface to give an efficiency of 40 per cent. In this case,

$$r_f = (96{,}000 \times 0.25) \div (64{,}000 \times 1.0) = 0.375.$$

Using Eq. (479) and Fig. 501, for $e = 0.4$,

$$A = \frac{96{,}000 \times 0.25}{8 \times 0.625} \times 0.35 = 1680 \text{ sq. ft.}$$

2. Suppose, with the above data, that the terminal temperatures of the flue gases and of the water are desired if an economizer of 5000 square feet heating surface is used. Then from Eq. (477),

$$n = (5000 \times 8 \times 0.625) \div (96{,}000 \times 0.25) = 1.042.$$

And from Eq. (481),

$$t_b = 900 - \frac{800\,(1 - \epsilon^{1.042})}{0.375 - \epsilon^{1.042}} = 304 \textit{ deg. fahr.}$$

From Eq. (482), $t_2 = 100 + 0.375 \times 596 = 324$ *deg. fahr.*

As a check, use Eq. (434), $A = q/U\theta_m$. As $\theta_a = 576$ and $\theta_b = 204$, so $\theta_m = 358$ deg. fahr.; then

$$A = (64{,}000 \times 224) \div (8 \times 358) = 5000 \text{ sq. ft.}$$

344. Economic Extent of Heating Surface.—(a) It has been seen that increasing the extent of heating surface in heat transfer apparatus increases the efficiency of the heating surface. Note carefully, however, that high efficiency does not necessarily mean high economy from the financial standpoint. Each added increment of surface improves the efficiency, but does so in less amount than did preceding increments and yet the cost is just as much per square foot of surface; and further, the enlarged surfaces may increase the expense for operation and maintenance of the apparatus and may add to the external losses. Hence, in a *heat recovery apparatus*, for example, there is an extent of surface beyond which (*a*) the annual saving effected by the improvement in heat absorption is more than offset by (*b*) the increase in the annual charges on the investment (including interest, depreciation, and upkeep) and in other expenses. This is shown graphically in Fig. 502, in which curve *b* is subtracted from curve *a* to get curve *c*. The highest point of the latter determines the **optimum extent of surface** A_0 giving the greatest annual saving that can be effected with the apparatus.

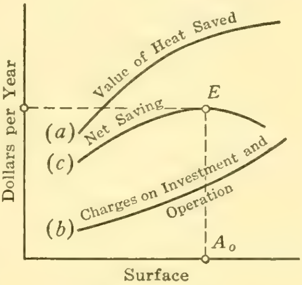

Fig. 502.

In general, the extent and efficiency of the surface for best economy is dependent primarily on (1) the overall coefficient of heat transmission, (2) the amount of time the apparatus is used during the year, (3) the cost of a unit of heat or of the fuel producing it, (4) the cost of the heat-transmitting apparatus per unit of surface, (5) the rates of interest and depreciation, and (6) the expense for operating pumps or fans for forcing the fluid through the apparatus. For a fixed value of (1) the smaller (2) and (3) and the greater (4), (5), and (6) are, the smaller will be the extent and efficiency of the surface for best economy.

(**b**) When many factors are involved in the determination of the economic extent of surface, empirical or cut-and-try methods, combined with graphical analysis, are often used. If, however, equations can be written to represent, on the one hand, the relation between the extent of surface, A, and the annual charge, Y', on the investment, and, on the other hand, for the relation between A and the annual cost Y'' for circulating the fluids and for other operating expenses, then the most economical area is determined by making

$$\frac{dY'}{dA} + \frac{dY''}{dA} = 0, \quad \text{or} \quad \frac{dY'}{dA} = -\frac{dY''}{dA}.$$

(**c**) The *velocity of flow* of the fluids has already been shown to affect very materially the overall coefficient of heat transmission, through its effect upon the film coefficient, but the velocity also affects the economic part of the problem because of the power required to produce the velocity. By using high velocities, the overall coefficient of heat transfer is increased, and therefore in a unit of time a given surface will transmit more heat per unit area and can therefore raise a greater amount of fluid through the desired temperature range; or, with a given quantity of fluid to be heated per unit of time, the amount of surface, and hence the cost of heater, can be reduced. But the increased velocity involves a correspondingly greater pressure drop through the apparatus, and thus an increased annual cost for operating the pumps or fans for forcing the fluid through the apparatus; this will partly or wholly offset the gain due to the higher rate of heat transmission. Hence for each set of conditions there is an **optimum velocity** which will give the greatest economy. The method of determining this velocity, v_0, is indicated in Fig. 503.

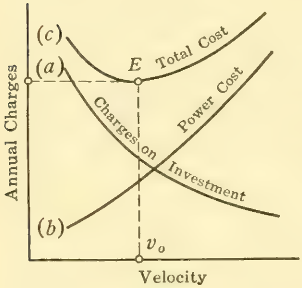

FIG. 503.

(d) When a *heat exchanger* is used to alter the temperature of one or both fluids through a definite range with forced circulation, it is desirable to use the smallest diameter (D) of tube practicable (allowing for use of tube cleaners, ease of construction, etc.), for it has been shown that in general U varies as $1/D^m$.

For example, if in a liquid heat exchanger U varies [38] inversely as $D^{0.124}$, then if $D^{0.124}$ be multiplied by the cost of exchanger per square foot for different sizes of tubes, the products will be proportional to the cost per B.t.u. per hr. per deg. and will indicate the optimum size of tube.

Since the amount of surface required in such apparatus is inversely proportional to U, which in turn is directly proportional to the pressure drop p raised to some exponent n, the yearly investment charge for the apparatus is $y' = \text{const}_1 \cdot w/p^n$, in which w is the weight of fluid handled per unit of time. The annual pumping charge is $y'' = \text{const}_2 \cdot pw$. Differentiating these two quantities with respect to p and equating them to each other gives an expression from which the **optimum pressure drop** p_0, can be obtained, provided the values of the constants have been determined for the particular kind of apparatus under consideration. With p_0 and D known it is possible to determine for a fluid of known characteristics the value of U to use with given temperature range of fluid and mean temperature difference. The amount of surface, the number and length of tubes, and number of passes required in the exchanger can then be computed. Instead of basing the analysis on the optimum pressure drop, it can be based on the optimum velocity of flow.

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[38] See "General Heat Transfer Formulas for Conduction and Convection," by E. R. Cox, Trans. A.S.M.E., 1927–28, Pet–50–2, p. 13.

(**e**) There are, of course, other methods that may be applied in the solution of such problems. The foregoing, however, will serve to indicate in a general way the considerations involved. As each class of apparatus has distinctive conditions to fulfill, further consideration of these matters will be deferred until such apparatus is discussed in detail in later chapters.

345. Determination of Economical Thickness of Insulation.—(**a**) The increasing demand for higher fuel economies in all industries has made the choice of heat insulation of great importance. The high temperatures of superheated steam in power plants, and the high temperatures of fluids in many industrial processes may cause large heat losses from bare metal surfaces. For example, a bare 10-inch pipe carrying superheated steam at 700 deg. fahr. with surrounding air at 80 deg. fahr. will lose about 9800 B.t.u. per hour per lineal foot of pipe. At a fuel cost of $0.35 per 1,000,000 B.t.u., 200 feet of such uncovered pipe would cause a loss of $6000 per year. With a suitable pipe insulation, the heat loss may amount to only 4 or 5 per cent of the loss from a bare pipe. Therefore, a careful selection of heat insulation is justified.

Each additional inch of insulation thickness installed costs practically the same, but pays a decreasing return on the initial investment, because the first few inches of insulation are the most effective in stopping the heat loss. The most economical thickness of insulation is that thickness which gives the lowest sum of the annual cost of insulation plus the value of the heat lost.

(**b**) The principal factors which affect the choice of insulation thickness are:

1. The amount of heat lost from the bare surface per hour.
2. The hours of operation per year.
3. The value of the heat.
4. The difference in temperature between the fluid carried and the surroundings.
5. The cost of the insulation.
6. The thermal conductivity of the insulation.
7. The fixed charges on the investment.
8. The shape of the surface, i.e., the diameter, if a pipe.

Experimental investigations of the rate of heat transfer from bare surfaces are by no means complete, but a few test results are available

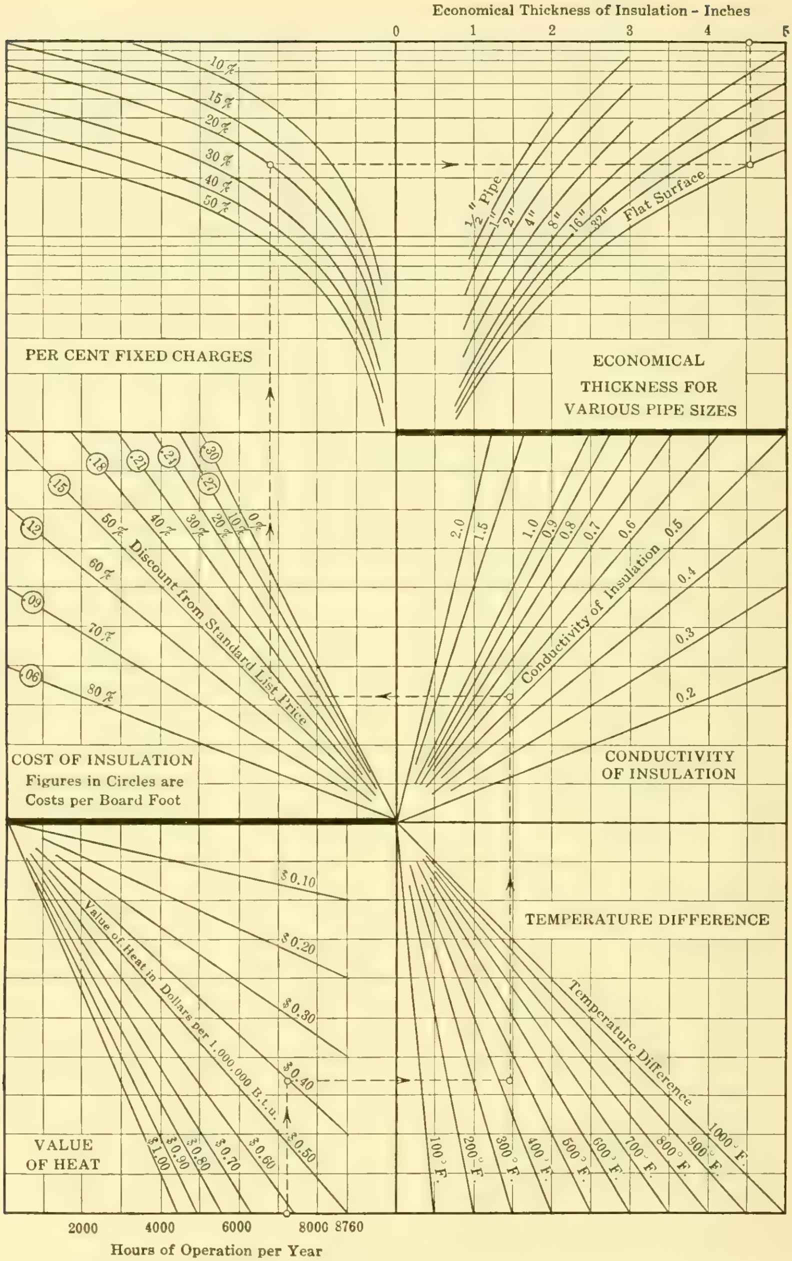

Fig. 504.—Chart for Determining the Economical Thickness of Insulation.

in the literature.[39] By means of the general principles of heat conduction given in Sect. **333** of this chapter, if the numerical values of the controlling factors are known, the economical thickness of insulation may be determined. Such a calculation is sufficiently laborious and is needed frequently enough to warrant a graphical solution such as that in Fig. 504 by L. B. McMillan.[40]

This chart is based upon rational equations[41] developed by Mr. McMillan, and its method of use is clearly indicated by following the dotted line starting at the bottom with the hours of operation per year.

[39] Trans. A.S.M.E., 1926, Vol. 48, p. 1290; Trans. A.S.M.E., 1922, Vol. 44, p. 299; Trans. A.S.M.E., 1915, Vol. 37, p. 941.

[40] Mechanical Engineering, May, 1929, p. 349.

[41] See Trans. A.S.M.E., 1926, Vol. 48, p. 1269.

ADDITIONAL REFERENCES

A.S.H.&V.E.—The Guide.

A. S. Refrigerating Engineers.—Refrigeration Data Book.

Badger, W. L.—Heat Transfer and Evaporation. Chemical Catalog Co.

Fishenden, M., and Saunders, O. A.—The Calculation of Heat Transmission. H. M. Stationary Office (London).

Grober, H.—Die Einführung in die Lehre von der Wärmeübertragung. Julius Springer, Berlin.

Grober, H., and Erk, S.—Die Grundgesetze der Wärmeübertragung. Julius Springer, Berlin.

Hausbrand, E.—Evaporating, Condensing, and Cooling Apparatus. D. Van Nostrand Co., N. Y.

Houghton, F. G.—Heat-Transfer Rates (Heating, Ventilating and Air-Conditioning). Mech. Eng., March 1934, p. 144.

King, W. J.—The Basic Laws and Data of Heat Transmission. A serial in Mechanical Engineering, 1932, pp. 190, 275, 347, 410, 492, 560.

Knaus, W. L.—Heat Transfer Rates in Refrigerating and Air-Cooling Apparatus. Mechanical Engineering, May 1934, p. 283.

Heat Transmission in Cooling Air with Extended Surfaces. Refrig. Engineering, Jan. 1935, pp. 23–26 (serial).

McAdams, W. H.—Heat Transmission. McGraw-Hill Book Co., New York.

Perry, J. H.—Chemical Engineers' Handbook, Section 7—Heat Transmission. McGraw-Hill Book Co., New York.

Schack, A. (Transl. by Goldschmidt and Partridge).—Industrial Heat Transfer. John Wiley and Sons, New York.

Sherman, R. A.—Radiation from Luminous and Non-Luminous Natural Gas Flames. Tr. A.S.M.E., 1934, IS–56–1, p. 177.

Burning Characteristics of Pulverized Coal and the Radiation from Their Flames. Tr. A.S.M.E., 1934, FSP–56–6, p. 401.

Chilton, T. N., Colburn, B. P., Genereaux, R. P., and Vernon, H. C.—Heat-Transfer, Design Data and Alignment Charts. Tr. A.S.M.E., 1933, PME–55–2, p. 7.

CHAPTER XXVII

FUELS

346. Fuels.—(**a**) In the broadest sense *fuel is any material which can be made to combine with other material in such a way as to liberate heat.* In the commercial sense, however, fuel is any material the greater part of which can be made to combine with *oxygen*, usually from the air, so as to liberate heat, and which is purchasable at such a price that its use will yield a profit.

(**b**) Fuels may be **solid, liquid,** or **gaseous.** The principal **natural fuels** are coal, lignite, peat, wood, crude petroleum, and natural gas. **Prepared or manufactured fuels** include: coke, charcoal, briquetted and pulverized solid fuels, tar, petroleum distillates and residuums, alcohols, liquid fuels formed by hydrogenation and cracking processes, and various kinds of manufactured gases. There are also certain kinds of **waste material** such as municipal refuse, sawdust, hogwood, bagasse (crushed sugar cane), gas from blast furnaces and coke ovens, and other manufacturing by-products which have fuel value.

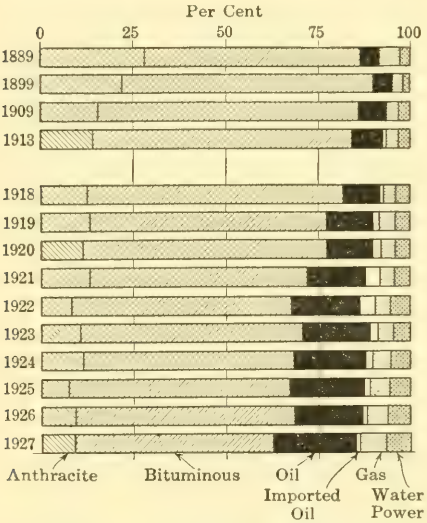

Fig. 505.—Per Cent of Total Energy Supply of the U. S. Derived from Various Sources, from 1889 to 1927. (Dr. Geo. H. Ashley, in "Combustion," June, 1930.)

(**c**) The percentages of the total energy supply of the United States derived from various mineral fuels and water power over the period from 1889 to 1927 are shown in Fig. 505, from which the relative importance of the different sources of energy can be readily seen.

347. Origin of Coal.—(**a**) Wood and other vegetable matter is composed principally of cellulose ($C_6H_{10}O_5$), water, various inorganic salts, resinous compounds, and generally a small amount of ash. Coal is composed of the elements C, H, O, N and S, together with ash and moisture, and is commonly believed to have been formed by the slow transformation of such organic matter into the mineralized state. It is assumed that beds of vegetable matter were first converted into peat by bio-chemical decay while under water, and subsequently, becoming deeply covered with silt and other material, they gradually changed in physical and chemical composition until they finally became coal. During the decaying process the mass became richer in carbon, which enters but slowly into other combinations, and became lower in oxygen and hydrogen, which react more quickly and apparently form gaseous combinations which leave the bed. The extent of these changes is dependent mainly on time, measured in geological ages, on temperature, and on the pressure, depth, and porosity of the overlying material. In general the greater the age, the more nearly will the state of mineral graphite be approached. Other factors that may affect the character of the coal formed are the kind of vegetable matter involved—e.g., the amount of gums, waxes, or resins contained—and the physical and chemical changes which may have been brought about in the early period by the bio-chemical actions of micro-organisms and fungi.

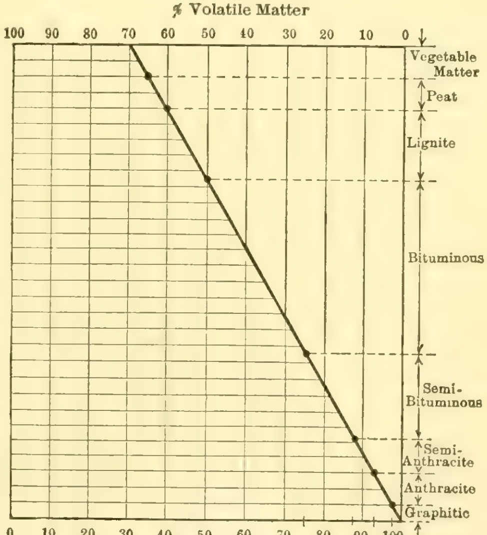

FIG. 506.

(**b**) The *combustible* part of coal consists principally of the *combustible volatile matter*, or the part that is released upon heating dry coal to a high temperature, and the *fixed carbon*, which remains after such treatment. As the formation of the coal progresses, the percentage of volatile matter decreases and a corresponding increase in fixed carbon takes place. The transformation of the combustible part is pictured in a very general way in Fig. 506, which shows the gradual loss of volatile matter and consequent gain in fixed carbon as the process proceeds. The divisions shown along the right side of the diagram seem to indicate well-defined demarcations between adjacent classes; but in reality the groups often blend into each other and overlap. The diagram is simply for illustration and should not be used otherwise. Considering the *complete dry fuel*—which is made up of the combustible, moisture of combination, and ash—as the transformations progress the inherent moisture content becomes proportionately less, decreasing from the high percentage in the original vegetable matter to almost nothing in the last stages of development; and, accompanying this change, the ash percentage naturally becomes greater.

348. Ultimate and Proximate Analyses and Heat Values of Coal.—(**a**) The only desirable chemical elements in coal for combustion purposes are the carbon and the hydrogen; but this fuel also contains oxygen, nitrogen, sulphur, and refractory material (ash), all of which may be considered as *impurities*. The various combinations into which the carbon, hydrogen, and oxygen are united are very complex and affect the physical characteristics of the fuel and the performance under service conditions; and any carbon or hydrogen that is already combined with oxygen in the coal as CO_2 and H_2O, respectively, is not available for combustion. Hence a chemical analysis that merely gives the percentages present of the elements C, H, O, etc., is not usually sufficient (*alone*) for determining the suitability of a coal for use under prescribed conditions. Nevertheless, such an analysis is always valuable for analytical purposes, for it reveals much information regarding the value of the fuel, and is useful in investigating the combustion processes and in determining what should be the composition of the exit gases when the coal is burned properly.

The composition of a coal can be determined in the laboratory by means of (1) an *ultimate analysis* and (2) a *proximate analysis*; and the heating value of the material can be measured by means of a *fuel calorimeter test*. Detailed consideration of the laboratory test-methods involved will not be undertaken in this text,[1] the discussion here being limited to general principles.

(**b**) Coal from the mines is somewhat wet, and it may lose some of this *free moisture* (or *hygroscopic* or *extraneous water*) during transportation and storage, or it may acquire more if exposed to rain. The ultimate and proximate analyses may be expressed on the basis of (1) *fuel as fired*, or "fuel as received," (2) *dry fuel*, or "fuel, moisture free," (3) *combustible*, or "fuel, moisture and ash free," and (4) "fuel, moisture, ash, and sulphur free," taken as 100 per cent or unity. The "dry fuel" and " combustible " still contain the *water of combination*, which is the union of H and O inherent in the fuel mass in its geological formation. The combustible includes the volatile matter, although some of this latter may be inert.

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[1] For test methods, see "Experimental Mechanical Engineering," Vol. I, by Diederichs and Andrae, published by John Wiley & Sons; and the A.S.M.E. Test Code for Solid Fuels.

(**c**) The **ultimate analysis** of a solid fuel consists of determining the weight percentages of the ultimate constituents (C, H, O, S, N, ash, and moisture) of a representative sample. The chemical determinations can be carried out satisfactorily only by trained chemists familiar with the methods. In some cases the total hydrogen and oxygen are reported as including the H and O in the hygroscopic moisture, but it is usually preferable to state the moisture as a separate item and not include its elements in the chemical analysis.

(**d**) The **proximate analysis** determines only four items: (1) *moisture*, (2) *ash*, (3) *fixed carbon*, and (4) *volatile matter*, of which the first two are identical with those in the ultimate analysis. This analysis requires less equipment than the former and can be made by any engineer, in accordance with the Test Code for Solid Fuels, although an accurate proximate analysis requires considerable skill. For many purposes the proximate analysis is sufficient, and it is the one most often made. It does not divide the volatile matter into its component parts. The so-called moisture content is an arbitrary quantity obtained by determining the weight lost by the sample when heated to a temperature of 220 deg. fahr. for an hour or so. This moisture should not be confused with the air-drying loss that results when a sample is brought to a condition of equilibrium with respect to the moisture in the air in the room. The "volatile matter" is determined in a similar manner but with the fuel at a "red to white heat" and with air excluded. The ash is the residue after complete combustion. The "fixed carbon" content is given by the difference between 100 per cent and the sum of the other three quantities. This "carbon" is not necessarily the "elementary" carbon of the original fuel, as carbon may be deposited from the volatile hydrocarbons during the process of driving off the volatile matter, and as this residue may also contain small amounts of H, O, N, and ash-forming constituents, and about half of the original sulphur content.

(**e**) Assuming that all of the oxygen in the coal analysis is in combination with hydrogen in the form of H_2O, and letting the symbols represent **weight percentages**, then, since water contains 8 parts of oxygen and one part of hydrogen by weight, the *free* or *available hydrogen* percentage equals $(H-O/8)$; and the *combined moisture* percentage is $1\frac{1}{8}O$. The *total moisture* is the sum of the combined moisture and the free moisture (as determined from the proximate analysis). Moisture adds to freight-handling charges, and sometimes also to the cost of preparing powdered coal; it reduces the heating value per pound of fuel purchased and increases the furnace losses.

(f) **Sulphur** is also often included in a proximate analysis, as well as in the ultimate, because its presence in the fuel is usually objectionable. It can be determined by analyzing the washings from the bomb calorimeter used in obtaining the heating value of the coal, or by using other methods. Although this element has a small heating value, its combustion products, when combined with moisture, form undesirable corrosive acids which may attack the metallic parts of economizers, air-preheaters, flues, and steel stacks; it impairs the quality of coke used for metallurgical purposes, and it usually affects detrimentally the fusibility and clinkering of the ash.

Nitrogen, though of no heating value, is of economic importance because of the ammonia formed from it as a by-product in coke manufacture and in carbonizing processes.

(g) The true **higher heating value** per pound of fuel is obtained by burning a sample in a fuel calorimeter. When the ultimate analysis is known, the H.H.V., in B.t.u. per pound of *combustible,* can be determined approximately by *Dulong's Empirical Formula,*

$$\text{H.H.V.} = 14{,}500C + 62{,}000(H - O/8) + 4{,}000S, \quad . \quad (498)$$

in which the symbols represent the weights of the elements per pound of combustible. The H.H.V. per pound of *fuel* (dry, or as fired) is found from the same equation by substituting for C, H, O, and S their weights per pound of *fuel* (dry, or as fired). The results are approximate because the formula does not take account of the heats of dissociation and similar obscure phenomena occurring during the combustion reactions. There are other empirical equations for the H.H.V., such as those of Evans, given in Table XXX. The heating value can also be determined approximately by plotting the composition of the fuel on charts, such as Fig. 509, which contain isocalorific lines.

In this country the higher heating value of the fuel is always used in connection with boiler tests, whereas in Europe the lower heating value is commonly employed, hence test results from the two sources are usually not directly comparable. The lower heat values give boiler efficiencies from 2 to 9 per cent larger than those obtained by using the H.H.V., depending on the percentage of hydrogen in the fuel.[2]

(h) In addition to determining the percentages of the various elements present in the fuel, the scientist desires to know what compounds the elements form and how they affect the properties of the fuel itself.

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[2] Kreisinger, "High and Net Heat Values of Fuel in Combustion," Aug. 1930, p. 25.

Methods of "rational analysis" to determine such factors are being investigated.[3]

349. Constitution, H.H.V. and Classification of Coal.[4] — (a) Various classifications of coal are needed to meet the diverse requirements of the scientist, the engineer, the user, and the trader, each class having many different points of view as to the information desired, hence the difficulty which is being encountered by the American Engineering Standards Committee in its efforts to devise a simple, uniform, and adequate system of classification. The two basic considerations are, *first*, the intrinsic chemical and physical properties of the coal due to its constitution, origin, and degree of metamorphism, and *second*, the purposes for which the coal can be used to good advantage. A few of the present methods of classifying and comparing coals follow.

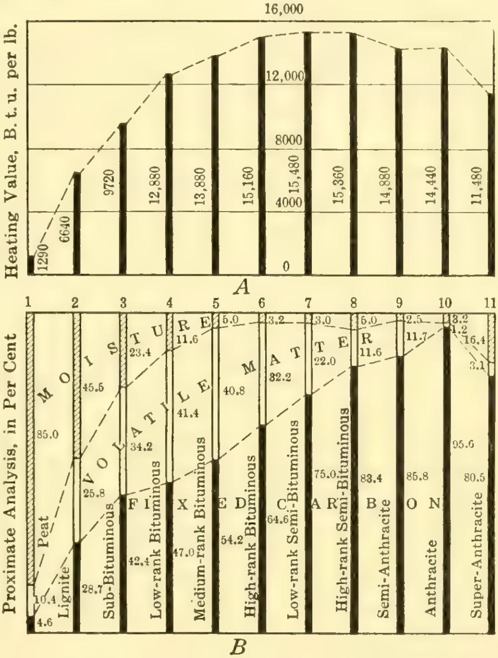

Fig. 507.—Diagram Showing the Effects of the Progressive Metamorphism of Vegetable Matter from Peat to Super-anthracite. Proximate Analyses and Heating Values on Ash-free Basis. (After M. R. Campbell.)[4]

(b) The various **ranks** assigned to American coals in the progressive order of development are (1) *peat*, (2) *lignite*, (3) *sub-bituminous*, (4) *bituminous*, (5) *semi-bituminous*,[5] (6) *semi-anthracite*, (7) *anthracite*, (8) *super-anthracite* (or *graphitic*) coal. The proximate analysis is sufficient to differentiate the ranks ranging from bituminous to anthracite, but for identifying sub-bituminous coals and lignite a knowledge of the physical characteristics is also essential. These characteristics are given in Table XXVII, in which FC/VM is the ratio of fixed carbon to volatile matter. The average ash-free proximate analysis on the percentage basis, and the calorific value of dry, ash-free combustible matter, in B.t.u. per pound, are given in Fig. 507. The upper diagram shows a progressive increase in calorific value from peat to low-rank semi-bituminous and then a decrease due to elimination of hydrogen. The progressive changes in moisture, volatile matter, and fixed carbon are clearly shown in the lower diagram. But there are no sharp boundaries between the ranks, not all coals with the same proximate characteristics are identical in properties, and the composition of the volatile matter varies with the composition of the coal; hence a more comprehensive differentiation of coals may be obtained on the basis of the ultimate analysis, more particularly in terms of the relative proportions of C, H, and O in the fuels free from ash, sulphur, and nitrogen.

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[3] See "Fuels and Their Combustion," by Haslam and Russell.

[4] Reference: "Constitution and Classification of Coal," by A. C. Fieldner, Ch. Chemist, U. S. Bureau of Mines. Trans. A.S.M.E., 1927–28, FSP 50–51.

[5] The term "semi-bituminous" is an unfortunate but long-established misnomer; it applies to a coal in a class *above* the bituminous, whereas the strict meaning of the term would indicate a lower position.

TABLE XXVII

Chemical and Physical Characteristics of Various Ranks of Coal (Fieldner)

Rank	Chemical Characteristics		Physical Characteristics
	Approximate Moisture,* Per Cent	Fuel Ratio FC/VM	
Peat	80–90	....	....
Lignite	30–45	....	Brown; clay-like or woody in appearance; slacks on exposure.
Sub-bituminous	12–30	....	Black; disintegrates on exposure, but less rapidly than lignite.
Bituminous	3–15	3 or less	Little or no slacking on exposure to weather.
Semi-bituminous	3–6	3–6	Friable; burns with little smoke.
Semi-anthracite	3–6	4–10	Hard; burns with very little smoke.
Anthracite	2–3	Over 10	Hard; high spec. grav.; smokeless.
Super-anthracite	2–13	Over 10	Resembles graphite.

* Mine sample. Ash-free.

(c) Considering only the carbon, hydrogen, and oxygen content of the fuel, with C + H + O = 100 per cent, the percentages of these elements in coals of different ages, and in the vegetable matter from which they are formed, average about as given in Table XXVIII.

TABLE XXVIII

	C	H	O
Cellulose	44.5	6.1	49.4
Wood	50.0	6.0	44.0
Peat	60.0	6.0	34.0
Lignite	70.0	6.0	24.0
Bituminous	82.0	5.5	12.5
Semi-bituminous	92.0	5.0	3.0
Semi-anthracite	94.0	4.0	2.0
Anthracite	95.0	3.0	2.0

If these percentages are plotted on trilinear coordinates,[6] as in Fig. 508, the resulting graph, **Ralston's curve,** indicates the alterations that occur in the relative proportions of these constituents as the transformation progresses. The curve so obtained lies in the sectioned region in the lower left corner of the chart, hence only that part need be used. Each of the different kinds of coal varies somewhat in composition, but the corresponding points for all except cannel coals fall not far from the graph of averages. By plotting thousands of analyses, O. C. Ralston determined the general trend of the curve,[7] as in Fig. 509, and found the zones commonly occupied by the different coals, the boundary lines between the fields being fairly sharp. The chart shows in a striking manner the rapid increase in carbon and decrease in oxygen in progressing from wood and lignites through to graphite. The older the coal, in the geological process of formation, the farther it is to the left on the figure. The division lines between the various fields are not perpendicular but are oblique across the band of dots. To illustrate the manner of reading the chart: for

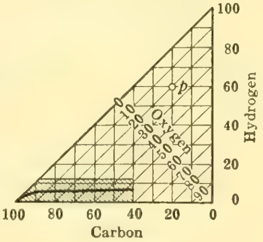

FIG. 508.

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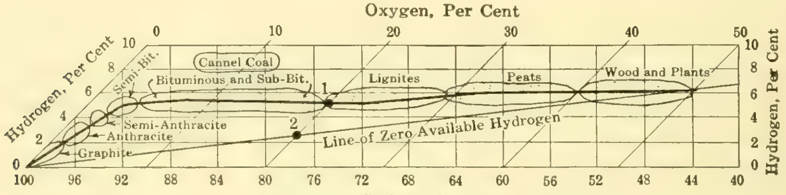
[6] It is a property of a triangle that if each side in turn is taken as the zero base and the perpendicular to the opposite corner is divided into percentages by lines parallel to the base, 100 per cent being at the corner, then for any point, such as *p* in Fig. 508, the coordinates with respect to the three sides give the percentages of the three components represented, their sum being 100 per cent.

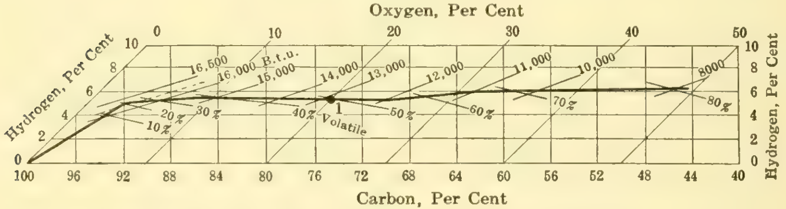
[7] This figure is redrawn with modifications from a much larger chart on which thousands of analyses were plotted and which was published in Technical Paper No. 93 of the U. S. Bureau of Mines, entitled "Graphic Studies of Ultimate Analyses of Coal," by O. C. Ralston, and from Fig. 33 of Bureau of Mines Bulletin No. 133, "Combustion of Coal and Design of Furnaces," by Kreisinger, Augustine and Ovitz.

a lignite located at point *1* the composition is 74.8 C + 5.2 H + 20 O = 100 per cent.

Assuming that all the oxygen in the analysis is combined with some of the hydrogen, forming H_2O which appears combined with carbon in carbohydrates, $C_x(H_2O)_y$, the oblique line of zero *available* hydrogen is drawn. The percentage of hydrogen available for combustion is the difference between the total H for any analysis (such as that for point *1*) and the amount in the carbohydrate having the same percentage of O (as given at point *2*). It is seen that the percentage of hydrogen available is zero for vegetable matter, increases to a maximum in the semibituminous region, and then decreases as the graphitic state is approached.



(*a*) GROUPING OF COALS, (C+H+O = 100%)



(*b*) ISOCALORIFIC AND ISOVOLATILE CURVES

FIG. 509.—Ralston's Chart.

(d) On his chart Ralston also drew lines through points of equal calorific value and through those of equal percentages of volatile matter, thus obtaining *isocalorific* and *isovolatile lines*, a few of which are shown in the lower chart (*b*) in Fig. 509. Progressing from right to left along the curve of average analyses in this chart, it is seen from the calorific lines that the heating value per pound of combustible matter[8] increases at first, reaches a maximum with semi-bituminous coal, then diminishes to the value for carbon alone; and from the isovolatile curves it is observed that a continual decrease in the percentage of volatile matter takes place throughout the whole range of geological formation.

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[8] Although not so stated, the heating values on the original charts are apparently on this basis, i.e., they are probably for coal that is dry and free from ash.

If the proximate analysis (including the sulphur percentage) and heating value of coal be known, then the percentage of volatile matter and the H.H.V. (when reduced to the basis on which the chart was constructed) can be used to locate a point with respect to the isocalorific and isovolatile lines. The position of this point then determines the classification of the coal and its approximate composition as to C, H, and O. Assuming a nitrogen content, usually about 1.25 per cent, data are then available for determining fairly closely the ultimate analysis and heat value of the coal in terms of fuel-as-fired, dry-fuel or combustible, whichever is desired.

(e) Briefly, Ralston's chart shows that as the geological transformations progress from vegetable matter to anthracite, (1) the carbon content increases, (2) the oxygen decreases, (3) the volatile matter becomes less, (4) the total hydrogen remains nearly constant until the semibituminous stage is passed, (5) the available hydrogen increases to a maximum at that stage, and (6) the heating value of the combustible also becomes greater until the same point is passed.

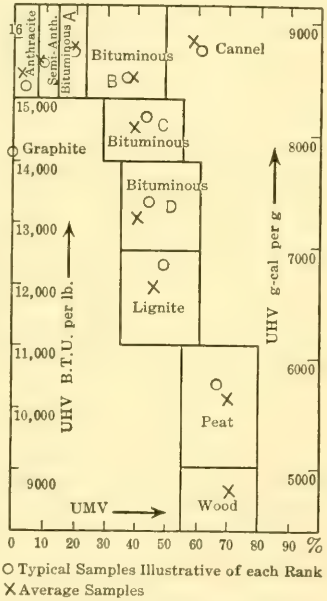

Fig. 510.—Parr's System for the Classification of Coal.

(From International Critical Tables)

(f) In **Parr's**[9] **system** of classification the percentage of volatile matter (U.V.M.) and heating value (U.H.V.) per lb. of "unit" coal are used as coordinates, as in Fig. 510. Unit coal is the portion free from moisture and ash and corrected for sulphur and water of hydration of shaly matter present. On this chart the coals in the different groups fall rather closely together, and within the arbitrarily located dividing lines.

(g) Coals are also classified in many other ways, such as: according to geological age; source, i.e., mine, vein, district, field, region, or coal province; ratio of components, e.g., O/H, C/H, FC/VM, etc.; use, e.g., for blacksmithing, manufacture of gas or coke, domestic heating, steam generation, distillation of by-products, or conversion into oil; caking or disintegrating qualities in service; weathering properties; relative hardness or other physical characteristics or properties; size, etc. Although there are no strict definitions as to what constitutes anthracite, bituminous, or other coals which bear the names used customarily in the coal trade, each is expected to have certain well-recognized characteristics.

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[9] S. W. Parr, "The Classification of Coal," Ind. and Eng. Chem., Vol. 14 (1922), p. 919.

350. Coal Fields and Resources of the United States.[10]—(a) Of the coal reserves of the world, this country apparently has over one-half, the more readily accessible portions of which it is estimated would make a cube with edges about ten miles long, the volume being 1000 cubic miles. But this supply is being depleted at an extremely rapid rate, as indicated in Fig. 511, the consumption in 1929 being 560 million tons, or about 2/10 cubic mile. The exhaustion of the tonnage remaining in the ground in some of the great producing fields in the east is within sight. The Pittsburgh bed may last but a single generation, the formerly important Georges Creek "Big Vein" in this bed already being almost worked out. Of the estimated twenty-one billion tons in the Pennsylvania anthracite region, about four billion tons have been removed from the best veins. Rational use, instead of prodigal waste and indiscriminate application of the finest coals, is becoming of more importance because of rising price and restrictions in supply. The power industry is making great gains in the conservation of fuel and in the reduction in the cost of fuel per unit of energy delivered. The distribution of use of bituminous coal in this country in 1927, on the percentage basis, according to the Census Report of that year, was about as follows: (1) locomotives, 24; (2) manufacturing industries, 20; (3) domestic uses, 20; (4) coke ovens, 15; (5) electric public utilities, 8; (6) iron and steel works, 5; and (7) other uses, 8. The annual consumption of coal by central stations in recent years is given in Fig. 512,

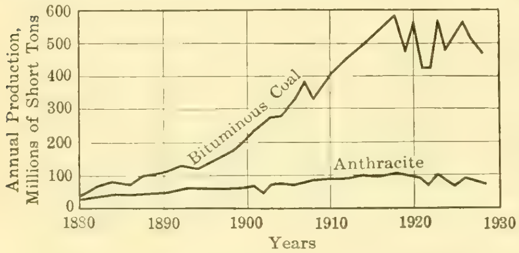

FIG. 511.—Annual Coal Production in the United States.

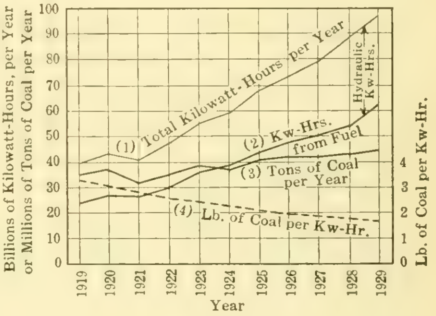

FIG. 512.

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[10] See Publications of the U. S. Geological Survey.

together with related information. It is seen that although the station output increased 128 per cent from 1921 to 1929, the fuel consumption was only 37 per cent more because the average weight of coal per kilowatt-hour was decreased from about 2.75 pounds to about 1.7, as shown by Curve (4).

(**b**) The main coal fields [11] in this country are shown in a very general way in Fig. 513, in which the average character of each deposit is indicated by the kind of hatching. There are numerous other fields of smaller extent scattered over the country, but they are not shown on this small map. In Rhode Island there is a little graphitic coal. Most

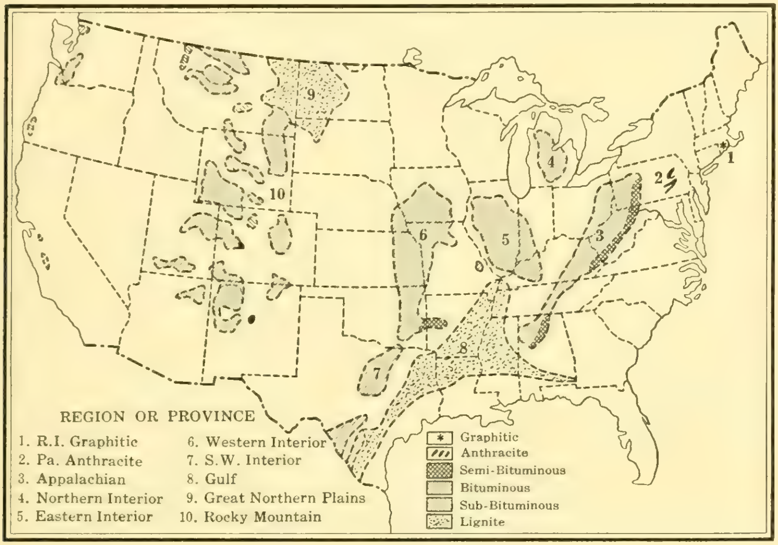

Fig. 513.

of the anthracite is found in beds of less than 500 square miles area located in eastern Pennsylvania. The principal deposit of semi-bituminous coal is about three hundred miles long by twenty wide and lies along the eastern edge of the Northern Appalachian Field. The bituminous coals extend from this deposit westward.

Starting with the graphitic coal in Rhode Island, broadly speaking, the farther west a coal is located the less advanced it is in the process of transformation. Thus, following Ralston's curve (Fig. 509) from left to right indicates in a general way the changes found in the composition of coal in going from R. I. to the West or Southwest. It is important to note, however, that there are many exceptions to these very general statements; for instance, a little anthracite coal is found in Colorado, New Mexico, and Arkansas, and some semi-bituminous in the latter state.

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[11] Professional Paper 100*a* and Bulletins 394 and 666*m*, of the U. S. Geological Survey.

(c) **Analyses of typical American coals** are given in Table XXIX. This table is very brief and is merely illustrative. An almost complete compendium of analyses of the coals of the United States is given in Bulletins 22, 85, 123, and 193 of the U. S. Bureau of Mines. Extensive tables are contained in M. E. handbooks and in reference books specially devoted to fuels and combustion.

TABLE XXIX

Analyses of Typical American Coals (Kreisinger) *

Kind of Coal	Proximate, Per Cent				Ultimate, Per Cent					Air Drying Loss in Weight, Per Cent	H.H.V., B.t.u. per Pound	
	Moisture	Volatile Matter	Fixed Carbon	Ash	Sulphur	Hydrogen	Carbon	Nitrogen	Oxygen		As Received	Moisture- and Ash-Free
R. I. graphitic coal	13.9	2.5	63.2	20.4	1.34	1.84	62.09	0.19	14.14	11.4	9,040	13,770
Penn. anthracite	3.43	6.79	78.25	11.53	0.46	2.52	78.85	0.77	5.87	2.5	12,782	15,030
Pocahontas coal	2.8	14.5	77.4	5.33	0.64	4.56	83.39	1.03	5.05	2.2	14,550	15,800
New River coal	2.94	20.11	73.1	3.85	1.12	5.04	82.60	1.46	5.93	2.1	14,582	15,625
Pittsburgh bed	3.39	31.79	56.46	8.36	1.05	5.07	74.42	1.39	9.71	2.4	13,699	15,522
Alabama coal	4.83	31.40	52.40	11.37	0.74	5.26	71.61	1.23	9.79	3.9	12,780	15,250
Illinois coal	12.39	36.89	41.80	8.92	3.92	5.85	61.29	1.00	19.02	8.4	11,399	14,500
Iowa coal	13.88	36.94	35.17	14.01	6.15	5.52	54.68	0.84	18.8	9.8	10,244	14,200
Kansas coal	4.99	32.68	49.36	12.97	4.28	4.98	67.34	1.08	9.35	1.3	12,242	14,900
Arkansas coal	2.91	12.65	66.93	17.51	3.12	3.60	70.88	1.17	3.72	1.7	12,312	15,440
Oklahoma coal	4.61	37.0	47.25	11.14	3.63	4.92	67.36	1.48	11.46	1.2	12,319	14,600
Colorado anthracite	3.0	3.0	86.5	7.5	0.69	2.67	83.20	1.48	4.48	1.1	13,500	15,100
Colorado bitum	4.06	34.48	52.84	8.62	0.65	5.34	71.18	1.24	12.97	2.4	12,888	14,750
Colorado lignite	23.07	31.20	35.60	10.13	0.21	5.77	47.69	0.64	35.56	14.5	8,030	12,000
Wyoming sub-bitum	22.38	31.85	39.42	6.35	1.16	6.32	52.25	1.19	32.73	8.1	9,247	12,900
Texas lignite	34.7	32.23	21.87	11.20	0.79	6.93	39.25	0.72	41.11	24.6	7,056	13,000
N. Dak. lignite	38.92	25.54	30.15	5.39	0.48	6.89	39.34	0.68	47.22	31.7	6,739	12,100
Florida peat	17.2	51.01	24.85	6.93	0.49	6.14	47.85	2.89	36.19	50.0	6,403	8,420
Alaska bitum	7.77	7.40	75.59	9.24	0.66	4.07	73.99	1.41	10.63	6.0	12,569	15,140
Alaska lignite	34.42	23.92	27.80	13.86	1.38	6.41	35.46	0.63	42.26	30.3	6,300	12,180

* From "Provide the Best Coal for Your Boiler Plant," Factory and Industrial Management, Dec., 1928.

(d) Calorific tests and chemical analyses are not as conveniently made by the engineer as are proximate analyses, and data as to the latter are more often available, as they are the tests more commonly made. Professor F. C. Evans[12] has shown that, knowing merely the percentage of volatile matter in the combustible of a coal and the locality from which the coal comes, it is then possible to determine quite accurately by means of empirical equations the heating value and the hydrogen and carbon content of the combustible matter. **Evans' Empirical Relations** are given in Table XXX.

TABLE XXX

EMPIRICAL RELATIONS FOR COALS OF THE UNITED STATES

Based on Proximate Analyses

Property of Coal Desired	States	Range of Volatile Matter in Combustible	Equation
Heating value of the combustible	Pa., O., W. Va., Md., Va., Ky., Ga., Tenn., Ala., Ind., Ia., Neb., Kan., Mo., Okla., Ark., and Texas	0-16 per cent	$H.V. = 14,550 + 7,810V$
		16-36 per cent	$H.V. = 16,160 - 2,250V$
		36 per cent up	$H.V. = 18,750 - 9,440V$
	Ill. and Mich.	All values	$H.V. = 16,062 - 3,830V$
Total carbon in the combustible	Penn., O., W. Va., Md., Va., Ky., Ga., Tenn., Ala.	0-36 per cent	$C = 0.943 - 0.242V$
		36 per cent up	$C = 1.095 - 0.663V$
	Ill., Ind., Mich., Ia., Neb., Kan., Mo., Okla., Ark. and Texas.	All values	$C = 0.953 - 0.362V$
	Col., Utah, N. Mex., Ariz., Wyo., Mont., Wash., Ore. and Cal.	36-60 per cent	$C = \frac{(H.V. + 7,544)(0.0099 + 0.0208V)}{737.5V + 200}$
	All states.	60 per cent up	$C = \frac{(H.V. + 7,544)(0.0099 - 0.0045V)}{11.3V + 200}$
Hydrogen in the combustible	All states but Ark.	4-16 per cent	$H = 0.013 + 0.225V$
		16 per cent up	$H = 0.0457 + 0.0206V$
	Ark.	All values	$H = 0.0327 + 0.056V$

H.V. = high heating value of the combustible in B.t.u. per pound.
V = fraction of volatile matter in the combustible.
C = fraction of total carbon in the combustible.
H = fraction of hydrogen in the combustible.
Combustible = Coal − Ash − Moisture.
= C + V = Unity.

NOTE.—The above table does not include lignites, peats, woods, or cannel coal except as noted in main parts of report.

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[12] Bulletin No. 3, Engineering Experiment Station of Cornell University, "Empirical Relations for Coals in the United States," by F. C. Evans.

Having given the proximate analysis and also the percentage of sulphur, which is often given with such data, and assuming a nitrogen content (about 1.25 per cent), the *ultimate analysis* of the *dry* fuel can be approximated by means of the formulas given in this table, in a manner similar to that which uses Ralston's Chart to obtain the same result (Sect. **349** (**d**)); and, if desired, these determinations can then be converted to the "as fired" or "as received" basis, if the probable moisture percentage is known or assumed. Even if the results thus obtained lack extreme accuracy, they may still be useful in various ways, such as in segregating losses that occur in coal-fired furnaces, or in establishing attainable standards of combustion performance which the operators should approach as closely as possible in actual practice.

(**e**) As a rule, the **sulphur** in eastern coals is from ½ to 2 per cent and in middle-western and western coals it is generally higher, some having as much as 5 to 8 per cent. **Nitrogen** ranges from about 0.75 to 1.75 per cent, the average being about 1.25 per cent.

351. Ash.—(**a**) The inert solid matter in coal consists of (1) the *fixed ash*, derived from the original vegetable matter and from the intimately mixed sedimentation, etc., and (2) the *free ash*, due to chunks of clay, shale, slate, pyrites, roof and floor fragments, etc. The first kind of ash cannot be separated readily from the coal; the second is reduced in amount by washing, screening, and hand picking. The percentage of free ash in small sizes of coal is normally greater than that in larger sizes because of greater difficulty in its removal. In commercial coals the ash percentage ordinarily ranges from three to fifteen.

Ash is undesirable because (1) it decreases the heat value of the fuel per unit weight, and (2) increases the cost of transportation, storage and handling of the coal per unit of heat produced; (3) the handling and hauling away of the ash itself adds to the operating expense, and a place must be provided for depositing this material; (4) this inert matter influences the combustion process—possibly by limiting it—and (5) may hinder the operation of stokers, or other combustion apparatus; also (6) *fly ash*, which is carried in suspension by the furnace gases when coal is burned intensively, and especially when the fuel is used in powdered form, may deposit on the furnace walls and cause their deterioration, (7) it may interfere with the furnace draft by bridging between boiler tubes, (8) deposits of it on heat transmitting surfaces may reduce the effectiveness of these surfaces, and (9) its discharge from the chimney may be a nuisance to the neighborhood. Further, (10) the ash content of coal is hard and abrasive, hence adds to the difficulty and expense of pulverization and increases the wear of the apparatus used.

Ash is not always wholly detrimental or useless, however, for with some grates and stokers a small amount of it is needed to form a protective coating on certain parts of the apparatus; it may have some commercial value as "fill" to make available swamps and other waste land, and it is extensively used as a component in the manufacture of cinder concrete.

(b) **Clinker** is formed by the mechanical adhesion of particles of the ash, or by the fusing of this material, thereby forming slag. When in a semi-fluid state it may spread out, choke air passages, and enclose large amounts of unburned fuel.

The formation of clinker depends mainly on (*a*) the composition of the ash, (*b*) the temperature of the furnace—as affected by the rate of combustion, amount of excess air used, etc.—and, (*c*) the extent of the direct exposure to the heat. As yet the effect of composition on the fusibility of ash is understood only in a general way, but it is known that the softening point of ash is lowered by the presence of magnesia, silica, lime, and iron compounds (particularly the sulphides). The fusion temperatures are determined by simple tests performed according to the A.S.M.E. Test Code for Solid Fuels.

(c) Coals and their **ash fusibilities** have been classified as follows:[13]

"Class 1.—Coals having ash which fuses at temperatures from 2600 to 3100 deg. fahr., and, in general, includes the anthracite coals and the lower and older bituminous beds. Ash from such coal is refractory, giving no trouble from clinkering.

"Class 2.—Coals having ash which fuses at temperatures from 2200 to 2600 deg. fahr. and, in general, includes the bulk of Pennsylvania bituminous beds. The ash from these coals is of medium fusibility.

"Class 3.—Coals having ash which fuses at temperatures from 1900 to 2200 deg. fahr. This ash is easily fusible and causes excessive clinkering trouble unless used properly."

With the intensive rates of combustion which are now prevalent in large plants, the fusibility of the ash is often a determining factor in the selection of the coal or the apparatus. It may be necessary either to choose coals having ash of character suitable for use with the apparatus available, or, in a new installation, to design the apparatus with respect to the qualities of the ash.

Important investigations of the properties of ash are now being conducted under the auspices of Engineering Societies and Government Bureaus.

[13] Bureau of Mines Bul. 209, "Fusibility of Ash and Coals of the U. S.," by Selvig and Fieldner.

352. Physical Characteristics of Coals.—(a) The classification of coal, and the suitability of the different kinds for various purposes are dependent to a large extent on the physical properties of the material, as well as on the proximate and ultimate analyses. Very brief descriptions of the physical characteristics of the principal kinds of coal follow:

(b) **Anthracite coal** ("hard coal") from Pennsylvania, when broken up, forms lumps that are hard, smooth, and shiny. It ignites slowly, burns only at a high temperature, produces little flame or smoke, and does not soften or swell. To obtain the best results with this fuel its lumps must be of uniform size. This coal is in great demand where smoke is not permitted; and, because of the limited supply, its cost per ton in sizes used for domestic heating is high. The very small sizes, formerly waste, are relatively cheap and are sometimes used in steam plants. They are burned on traveling-grate stokers, with forced-draft, the flame being very short and nearly transparent. Anthracite is sometimes burned in powdered form, but its pulverization is more difficult and expensive than that of the softer coals. *River coal*, used in a few plants, is small anthracite dredged from streams into which the mines run their refuse water. *Culm*, a refuse from screenings, and *bone* coal, containing so large a part of inert material as formerly to be unmarketable, are now also used in power plants. Arkansas and Colorado anthracite is softer than Pennsylvania anthracite.

(c) **Semi-anthracite coal** has less density, hardness, and luster than anthracite, but kindles and burns more readily, yielding greater heat, with little clinker and ash. It swells considerably, does not cake, but tends to split on burning and to waste through the grate. It comes chiefly from small areas in Pennsylvania, Virginia, and Arkansas and is little used in steam power plants.

(d) **Semi-bituminous coal** is a low-volatile coal which resembles semi-anthracite in appearance, but is softer. Under rough handling it reduces to slack which has higher heating value than the remaining lumps, which are usually held together by slate or other hard impurities. It has high calorific value and high available hydrogen; is low in moisture, ash, and sulphur; burns freely with little smoke, and is usually of the caking type, tending to form a crust impervious to air flow unless the fuel bed is constantly broken up. Stokers of the underfeed type which agitate the fuel bed are the most suitable ones for burning fuel of this kind. This coal is one of the best for steam generation and is often used as a domestic fuel. It burns on grates with a comparatively short flame, pulverizes easily, and in powdered form its *volatile matter* burns with a short, white, visible flame and without smoke, but the carbon residue burns slowly with a long *invisible flame* which necessitates using a comparatively large furnace volume. There may be a greater percentage of combustible lost in the flue dust than with other bituminous coals. Coals of this rank come from comparatively small areas known as the Pocahontas and New River Fields in Virginia and West Virginia, the Windber Field in Pennsylvania, the Georges Creek Field in Maryland, and the western Arkansas fields.

(e) **Bituminous coal** ("soft coal") comes from many regions and has widely varying properties. Although it often has about the same ratio of volatile matter to fixed carbon as the coals of lower rank, it is distinguished from them by the fact that *weathering* reduces its commercial value but little, as exposure affects it physically and chemically only slightly unless continued for years; and by the fact that it breaks up into prismatic fragments instead of into thin plates. It is the most widely distributed and extensively used fuel for steam generation. Bituminous coals are either *caking* or *non-caking.*

The *Eastern* or Appalachian coal field, including the *Pittsburgh* district, contains high-grade bituminous coal with volatile matter from 28 to about 35 per cent. This coal stands handling without breaking into slack and when burned on grates or stokers tends to fuse and form a mass of coke, hence underfeed stokers are the type most suitable for this coal. It does not pulverize easily, and in powdered form it burns with a rather long luminous flame. If the air supply and mixing in the furnace are inadequate, smoke may be produced. The ash fusion temperature ranges from about 2000 deg. to 3000 deg. fahr., but for most of these coals it is between 2200 to 2600 deg.

Illinois bituminous coals are distinctly laminar in structure, are moderately hard and stand handling well. They are *free burning,* i.e., burn without fusing, but if agitated, the ash, which is more fusible than that of most Appalachian coals, forms troublesome clinkers; hence this coal burns best when quiescent, as when used with traveling-grate stokers. This coal pulverizes rather easily, except when containing an excessive amount of pyrite, and in the powdered state burns with a rather long luminous flame.

Iowa bituminous coal resembles Illinois coal in structure and composition but generally has higher percentages of moisture and ash, hence its heating value is proportionately less. *Kansas, Arkansas,* and *Oklahoma* bituminous coals are also similar to Illinois coals, but usually contain less ash and less moisture. The ash of mid-western coals commonly has fusion temperatures between 2000 and 2400 deg. fahr.

(f) *Western* **sub-bituminous** coals are somewhat laminar in structure, black, and frequently have shiny surfaces when first mined, but when exposed to weather they assume a characteristic dead-black color.

They contain much moisture and generally are low in ash. Exposure to air reduces the moisture, and if prolonged, the lumps disintegrate into slack. They burn freely with a long yellow flame and are likely to produce smoke. These coals are easily pulverized and burn satisfactorily in that form.

(g) American **lignites** are brown and distinctly woody, frequently showing the original wood structure. When first mined they contain 35 to 40 per cent of moisture and are tough. Exposure to air causes them to lose moisture and to disintegrate rapidly into slack. In the fire, the lumps crumble into very small pieces, some of which may be carried unburned out of the furnace by forced draft, hence these coals can be burned on grates or stokers only at very low rates. When partly dried, to about 28 per cent moisture, they are easily pulverized and burned.

(h) **Peat** is seldom used as a fuel in the United States. When removed from the bog its moisture is high. It is sun-dried before transportation and may be compressed into briquets and dried to about 20 per cent moisture. It burns like lignite, but the cost of its preparation prohibits its extensive use.

(i) **Other Coals: Cannel coals** are bituminous ones which are very rich in hydrocarbons (from vegetable waxes and resins). The richer portions of such coals burn like candles, hence the name. They plot well above Ralston's curve and are used for enriching illuminating gas and for the distillation of oil. Coals for producing coke for metallurgical or other purposes, for high or low temperature carbonization, for making various kinds of gas, tar, oil, by-products, etc., in each case should possess certain desired properties which, however, cannot be considered in this brief discussion.

353. Coal Preparation and Products.—(a) In the preparation of coal for the market it may be *sized* by screening and *cleaned* by removing the extraneous impurities either by hand picking or by dry or wet mechanical processes (washing). Coal is burned raw, "as received," or is first crushed, dried, briquetted, powdered, or otherwise prepared. It is used for generating power, for supplying heat for industrial or metallurgical processes or for warming habitations, or for making material products such as coke, gas, oil, and numerous by-products. The form in which it is utilized depends on many factors, such as the properties and condition of the coal itself, purposes for which it is to be used, the kind of apparatus and process involved, etc.

(b) **Coke** is the principal product made from coal, about one-seventh of the bituminous coal mined being used for this purpose. It is generally made by the distillation of a suitable coal or mixture of coals.

Coking coals fall near the break in Ralston's curve and are those caking or non-caking kinds that by actual test yield coke of adequate purity as to ash, sulphur, and phosphorus, and of satisfactory size, strength, and structure for the uses to which it is put. On the percentage basis, the disposition of coke in 1924 was as follows: blast furnaces, 82.6; foundry, 6.6; domestic, 6.7; water gas manufacture and other uses, 4.1. *Coke breeze*, or fine refuse, is low-priced and sometimes is used in power plants. Dry coke is composed mostly of carbon and ash, with small amounts of volatile matter and sulphur.

(c) Coke has usually been formed from the coal by **high-temperature carbonization,** the destructive distillation being carried to temperatures of from 1800 to 2100 deg. fahr. either (1) in beehive ovens from which the gas is wasted, (2) in by-product ovens, with recovery of tar, oil, ammonium sulphate, motor benzol, etc., or (3) in gas retorts, with the production of gas as the prime consideration. The tar recovered is used as road and roofing material, as fuel, and as the source of many derivatives, such as dyes, solvents, disinfectants, drugs, perfumes, explosives, etc.

(d) **Low-temperature carbonization** of raw coal is receiving attention as a means of conserving valuable by-products and of producing a low-volatile, smokeless fuel or semi-coke. Many processes for this purpose are being investigated and such methods of treating coal may become of importance in power-plant engineering. The tar and by-products recovered at the temperatures employed, the maximum being seldom above and often below 1100 deg. fahr., differ from those obtained by the high-temperature process. The thin tar obtained may be cracked to yield a large percentage of motor fuel.

(e) By **briquetting** a disintegrated coal—i.e., compressing it (with or without tar or other binding material) into bricks, or briquets—a low grade, and perhaps valueless material, may be converted into a useful fuel that will stand weathering and handling. In this form the coal is more commonly burned for domestic purposes than for other applications. **Powdered fuel** is extensively used under boilers and in the metallurgical and cement furnaces. Many low-grade as well as high-grade coals are burned advantageously in this form. The preparation and use of powdered coal in power plants will be considered later. **Fuel amalgams** can be formed by combining a little oil with ground graphitic coal, or other solid fuel that is difficult to burn, and this material can be briquetted and baked, as in the Trent process. **Colloidal fuel** can be made by emulsifying powdered coal with oil, usually with the aid of a "fixateur," to maintain the solid matter in suspension. The product can be burned by the methods used with fuel oils. By **hydrogenation** processes the combustible part of coal can be converted into oil from which liquid fuels and other valuable products can be obtained. This part can also be converted wholly or partly into gas, accompanied by the production of useful by-products.

(f) With the increase in the cost of coal of the better grades and of petroleum and natural gas, due to growth in demand or decrease in supply, and with development of uses for by-products that are more valuable than the coal from which they are made, conditions may eventually become such as to make it an economic necessity to use for the generation of power only low-grade coal which has been subjected to previous preparation, coal from which valuable derivatives have first been extracted, or oil or gas which has been produced from coal by processes similar to those mentioned. There are many fruitful fields for investigations along such lines.

354. Storage Properties of Coal.—(a) Coal is stored to ensure an uninterrupted supply and to take advantage of fluctuations in price. Power plants and manufacturing concerns often carry a stock sufficient for a period, say from one to three months, hence the effect of storage on the properties of coal is often an important matter.

Anthracite is practically unchanged during storage; but, in general, with bituminous coals, the lower the rank the greater is the deterioration from exposure. Even at ordinary temperatures a slow oxidation of the latter coals occurs, and if the heat evolved is not dissipated it may cause spontaneous ignition. Oxidation before the coal is used lowers the heating value per unit of weight,[14] affects the firing properties, impairs the coking qualities, and may have an effect on the by-products recoverable. (1) Oxidation takes place most rapidly on freshly exposed surfaces such as those resulting when the coal is first broken out of the mine or when it is crushed; (2) it is accelerated with decrease in size of particles, since the surface area per unit weight is increased; and (3) its rate is dependent on the initial temperature. The dissipation of heat occurs by conduction and by convection through the air circulation. There is a balance between heat generation and dissipation with a resultant temperature above which spontaneous combustion will occur and below which it is avoided.

(b) **Spontaneous combustion** is a possibility if a temperature of 150 deg. fahr. occurs within a coal pile, and usually is a certainty after 180 deg. has been reached. A deep pile having a mat of fine particles on top, and a base of large lumps which admits air to the interior, does not allow air circulation and is an arrangement which courts spontaneous ignition; and this is the usual distribution of sizes that results when coal is dropped from a height onto a pile. Anything that tends to increase the temperature of the coal at the time of storing or thereafter, or that causes breakage, is a contributing factor. The oxidation of pyrites (FeS_2) in the ash, although it may not be of major importance, often produces an accelerated action because it liberates heat and splits the coal into smaller particles. This reaction requires the presence of water, hence it is most likely to occur in moist coals.

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[14] Slow oxidation increases the weight of a coal pile due to the oxygen added. Therefore, the heating value of the whole pile is not much affected, but that of the coal may be appreciably less per pound.

(c) **Weathering,** in addition (1) to its influence on spontaneous combustion, may have the following effects on coal: (2) The appearance may be changed due to oxidation, especially that of the sulphur content. (3) The *heating value* may be reduced from 1 to 3 per cent in five years in bituminous coals of low oxygen content, and from 4 to 6 per cent in coals of lower rank, the greater part of the loss, however, occurring shortly after mining. (4) The *size* is reduced by the physical and chemical changes due to exposure. (5) The *firing*, *coking* and *other properties* may be affected by these changes. In general, the *lower* the rank of the coal, i.e., the larger the original oxygen content, the greater is the deterioration from weathering; hence, in a way, weathering may be looked upon as the reverse of the deoxidation process which takes place in the formation of coal.

(d) The **storage** of coal should be so planned as to avoid the foregoing possible losses. Air should be either excluded from the bed or be allowed to circulate freely through it. Coal should be piled with the lumps and fine particles evenly distributed; and to effect this, deep piles may be built up in layers two or three feet thick, each layer being leveled and possibly compacted by means of tractors before adding the next. Storage during cold weather is preferable; combustible foreign matter should be eliminated; and external sources of heat (steam pipes, etc.) and air circulation, such as that around posts, through porous base, etc., should be avoided. Before storage, as long a time as possible should be allowed for preliminary seasoning. To provide for temperature determinations, pipes may be driven into the pile at ten-foot intervals, but these pipes should be closed so as not to act as chimneys and not fill up with water. Fires are generally found from five to eight feet below the surface on the flanks of the piles. Storage under water avoids spontaneous heating and causes negligible deterioration, but, as it is expensive, it is practicable in only a few instances.[15]

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[15] References:—Technical Paper 311, U. S. Bureau of Mines, "Factors in the Spontaneous Combustion of Coal," by O. P. Hood (1912), and Technical Paper 409, "Spontaneous Heating of Coal," by J. D. Davis and D. A. Reynolds (1928). The latter paper contains an extensive bibliography on the subject.

355. Coal Sizes.—Anthracite coal can be burned efficiently only when of uniform size; and, similarly, it is often advantageous with other coals to have uniformity. The size best adapted to any given purpose is dependent on such factors as the kind of product desired, the strength of draft available, kind of combustion equipment used, intensity of combustion required, etc. A given size of coal will pass through screen openings of one size and over those of the next smaller magnitude. The shape of hole, however, has a bearing on the amounts and true sizes removed, hence trade descriptions may be confusing; a $1\frac{1}{4}$ in. product from a bar screen will have a maximum size quite different from that from a screen with $1\frac{1}{4}$ in. square holes and still different from one with $1\frac{1}{4}$ in. round openings. No universal standard of coal screening is followed in commercial practice because of differences in the customs in the various districts, in the market conditions and in the character of coals. The sizes are about as given in Table XXXI. In coal *testing*, the size is

TABLE XXXI

COAL SIZES

Trade Name	Diameter of Opening through or over Which Coal Will Pass, Inches	
	Through	Over
Anthracite		
Broken	$4\frac{1}{2}$	$3\frac{1}{4}$
Egg	$3\frac{1}{4}$	$2\frac{3}{8}$
Stove	$2\frac{3}{8}$	$1\frac{1}{4}$
Chestnut	$1\frac{1}{4}$	$\frac{3}{4}$
Pea	$\frac{3}{4}$	$\frac{1}{2}$
*Buckwheat No. 1	$\frac{1}{2}$	$\frac{1}{4}$
†Buckwheat No. 2	$\frac{1}{4}$	$\frac{3}{16}$
‡Buckwheat No. 3	$\frac{3}{16}$	$\frac{3}{32}$, $\frac{1}{16}$
Culm	$\frac{3}{32}$, $\frac{1}{16}$	....
Eastern Bituminous		
Run of Mine	As mined	
Lump	....	$1\frac{1}{4}$
Nut	$1\frac{1}{4}$	$\frac{3}{4}$
Slack	$\frac{3}{4}$	....
Western Coals		
Run of mine	As mined	
Lump, 6-in	....	6
Lump, 3-in	....	3
Lump, $1\frac{1}{4}$-in	....	$1\frac{1}{4}$
Nut, 3-in	3	$1\frac{1}{4}$
Nut, $1\frac{1}{4}$-in	$1\frac{1}{4}$	$\frac{3}{4}$
Nut, $\frac{3}{4}$-in	$\frac{3}{4}$	$\frac{5}{8}$
Screenings	$1\frac{1}{4}$	....
Franklin Co., Ill., Std.		
Small egg (No. 1)	3	2
Stove (No. 2)	2	$1\frac{1}{4}$
Chestnut (No. 3)	$1\frac{1}{4}$	$\frac{3}{4}$
Pea (No. 4)	$\frac{3}{4}$	$\frac{1}{4}$
Carbon (No. 5)	$\frac{1}{4}$	....

* Buckwheat, † Rice, and ‡ Barley are other names applied, respectively, to these small sizes of anthracite coal.

determined according to instructions contained in the A.S.M.E. Test Code for Solid Fuels. The Code discourages the use of trade names and gives preference to designating the size of standard screen perforations through and over which the coal will pass.

356. Selection of Coal; Coal Specifications.—(**a**) The objective in coal selection is to choose the particular fuel that will produce the desired product, or products, satisfactorily and with the greatest economy, the main factors to be considered being (1) suitability, (2) availability, (3) dependability of supply, and (4) cost of the fuel. Before constructing a *new* plant for producing power or other product, a study should first be made of the fuels available in the locality; then the combustion and heat transmitting equipment should be selected or designed with respect to the fuel, or fuels, best fulfilling these four requirements. In choosing a coal for use with *existing* equipment, its performance in such apparatus must be carefully considered; possibly it may be more advantageous to redesign some of the equipment to use a fuel which will effect better results.

(**b**) Usually it is desired to have fuel available from more than one source of supply, and to have alternative lines of transportation, so as to insure continuance of operation, despite accidents and strikes at the mines, or on rail or water transportation lines serving the plant. The greater part of the cost of the coal delivered at a distance from the mines is made up of freight charges which must be paid not only on the combustible matter but also on the ash and moisture. A coal bringing a high price at the mine but also having a high heating value may possibly be cheaper to use than one that is low priced at the source but of low heating value. The final test is the amount of heat or other product produced per unit of cost (all operating costs being considered) provided the coal is otherwise satisfactory. Coal for a power plant should be selected not only on the cost per million heat units delivered, but on the cost of handling and disposing of the ash, cost of fire room labor, cost of carrying banked boilers, suitability for the draft conditions, forcing capacity of the apparatus, and responsiveness to load changes with it.

(**c**) The factors that influence the combustion efficiency are included mostly in the chemical and ultimate analyses of fuel. However, the physical properties of the material are also important. Whether a coal cakes, burns freely, or slacks when heated and whether it burns with long or short flame must be considered in choosing the method and equipment for its combustion, in designing the apparatus for the absorption of the heat produced, or in deciding on the suitability of the fuel for use with existing apparatus. The volatile content of a coal, although of high heating value, may prevent the use of that fuel because of limitations imposed by the design of the combustion chamber, or by local smoke ordinances. Information not only as to the amount, but also as to the fusibility, clinkering, and other properties of the ash may also be needed, for these characteristics may place limitations on the kind of apparatus with which the fuel can be used and on the capacity output obtainable per dollar invested in the equipment. The sulphur content may be an important factor.[16] The size of the coal that gives the best results, and the storage properties of the fuel should likewise be investigated.[17]

(d) There are many **methods of purchasing coal.** Some buyers rely entirely on the honesty of the coal company to continue to furnish at a fair price a quality of coal that has been found suited to their needs. Many coal producing and selling companies maintain dependable engineering service-organizations for the benefit of their customers. At the other extreme, extensive purchasers, including the federal, state, municipal departments, and large commercial concerns, often buy their coal on the basis of specifications.

In **specifications** the contract price is usually based on the heating value per unit weight of *dry* coal, as the content of extraneous moisture is largely beyond the control of seller and purchaser; and fair adjustment clauses are incorporated to cover variations from standard quality. When moisture-limiting clauses are included, the adjustments should be made on the basis of samples made at or near the mines. Since high ash percentage is very objectionable, penalty and bonus provisions are generally included in the specifications to cover variations in this content. As the analysis and heating value of dry and ash-free coal from a given mine remain substantially constant, some specifications are based on the ash content only, and hence do not require calorimetric determinations. The fusing temperature, and the limiting amounts of ash and sulphur tolerated are also often specified.

The difficulty and expense of securing representative samples[18] satisfactory to both parties, and of obtaining analyses of composition and calorific values, usually make it inexpedient for the small consumer to make his purchases according to specifications. In that case the selection of coal may be based on the name and reputation, price, general character, and results known to have been obtained in actual use of a coal; reliance may be placed on the advice of the service department of the seller, if he has one; or outside fuel-and-combustion specialists may be employed to guide in the purchase and use of the coal. Large purchasers can afford to have their own staffs of specialists and the necessary laboratories, and may buy according to specification. Coal purchased under government specifications is discussed in the U. S. Geological Survey Bulletins 339, 378 and 428, and in U. S. Bu. of Mines Bulletin 116.

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[16] U. S. Bu. of Mines Tech. Paper 436, "The Sulphur Problem in Burning Coal," by J. F. Barkley.

[17] Analyses of thousands of coals, including all parts of the United States, are contained in U. S. Bu. of Mines Bulletins 22, 85, 123 and 193; Bul. 23 gives the burning characteristics of many coals; and Bul. 209 is devoted to the fusibility of ash.

[18] For Standard Method of Sampling and Testing Coal, see A.S.M.E. Test Code for Solid Fuels, or A.S.T.M. Standards.

A comparison of bids can be made by adjusting all coals to the same ash percentage (using as standard the coal offered with the lowest ash content) and making allowance for the varying heating values by computing the cost per million B.t.u. of each coal offered. One method of ash adjustment is to assume a negative value of two cents per ton for 1 per cent excess of ash over the standard.

357. Wood.—(**a**) As wood is about half moisture when felled, it must be dried before it is of much value as fuel. Air-dried wood generally has from 15 to 20 per cent of moisture, about 50 per cent of carbon, from $\frac{1}{4}$ to 2 per cent of ash, and the rest is volatile matter largely inert. The heating value per pound of dry material is from 6600 to 9800 B.t.u.

When other cheap fuel is available wood is not generally used for power production. However, refuse from sawmills and other woodworking factories may be profitably utilized. *Hogged Fuel,* which is used extensively in some places, properly speaking is waste wood that has been passed through a "hog" which cuts or hammers it down to a uniform small size; but the term is also applied to small recoveries from the manufacture of lumber. With one "unit" of 200 cubic feet of this material, weighing from 2500 to 5000 pounds, it is possible to generate from 200 to 1000 kilowatt-hours of energy, the amount depending on the kind of wood, moisture content, and efficiency of the boiler unit. Experience indicates that higher furnace temperatures, by 200 or 300 deg. fahr., can be obtained with wood than are possible with coal.

(**b**) **Charcoal** is made from wood in much the same manner that coke is made from coal. It is ordinarily used in power plants only when it is the by-product of some local process, such as the manufacture of turpentine or wood alcohol. It may contain from 80 to 97 per cent of carbon, depending on the temperature and treatment used in carbonizing it. The heating value is about 12,850 B.t.u. per pound.

358. Municipal and Industrial Waste.—In cities and industrial centers there is a constant accumulation of combustible waste and in some cases this material is burned directly as fuel, or it is gasified in suitable apparatus and the resultant combustible gas used as fuel. Installations of this character are still rare but are of growing commercial importance.

359. Petroleum and Its Products.—(a) **Petroleum,** or *Crude Oil,* is a more or less viscous, dark brown, or greenish-colored liquid occurring in natural reservoirs in the earth's crust. It is a mixture primarily of various hydrocarbons which are liquid at ordinary temperatures and pressures, and which hold in solution numbers of other hydrocarbons that otherwise would be gaseous or solid under existing conditions; but as it comes from the wells it contains also small amounts of oxygen, sulphur, nitrogen, water in emulsion, and silt, all of which are undesirable components. In general, oils from one field are composed of the same hydrocarbons in about the same proportions, but each field has its own characteristic composition.

(b) Many of the more highly inflammable volatile components tend to distil off when the oil is brought to the earth's surface and is exposed to atmospheric conditions. These volatiles, some of which boil at temperatures as low as 80 deg. fahr., are usually distilled off progressively in a refinery. The distillation products most commonly used as fuels are naphtha, gasoline, kerosene, and fuel oil, given in order of decreasing inflammability and increasing density and distillation temperature.

(c) The most accepted theory as to the **formation of petroleum**[19] credits the origin of this natural fuel to prolific marine growths which, hundreds of thousands of years ago, became covered to great depths by deposits under which they decomposed into hydrocarbons. In many instances these deposits were compressed into the shale and limestone of to-day, which, by reason of their impervious formation and water-saturated nature, became cap rock confining the oil and dissociated gas. Disturbances left the oil-bearing strata in domes, folds, and slopes. Within the strata the petroleum was forced into cavities or porous sand strata, assuming positions of equilibrium under the gas pressure and the buoyant action of the water underneath. Heat from subterranean sources and pressure caused the liquids to form various combinations, the lighter ones and the gases escaping when possible, thus forming crude oils of different densities. Oils found adjacent to coal regions may possibly have been formed by the decomposition of vegetable matter.

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[19] "The Petroleum Industry," by Walter Samans, Mechanical Engineering, Apr., 1930, p. 421.

(d) Regardless of its source, appearance, or characteristics, petroleum and its products have **ultimate analyses** falling within the following narrow limits:

	Carbon	Hydrogen	O + N	Sulphur
Percentages:	83 to 87	11 to 14	0 to 7	0 to 4

But such analysis is of secondary consideration, since the value of the fuel depends mainly on the kinds of hydrocarbons which the carbon and hydrogen form within it, on the impurities contained, and on the physical properties, especially the density of the oil. In fact, equations for determining numerical values of many of the properties of petroleum at the given pressure and temperature can be written in terms of the density alone.

(e) The **density** of oil is given either in terms of *specific gravity* compared with water at 60 deg. fahr., in degrees A.P.I, or in degrees on the *Baumé* hydrometer scale, the measurements being made with the oil also at 60 deg. fahr. The *American Petroleum Institute degree* and the specific gravity have the following standard relation:

$$\text{Degrees A.P.I.} = \frac{141.5}{\text{sp. gr., }60/60^\circ\text{ F.}} - 131.5 \quad . \quad . \quad (499a)$$

or

$$\text{Sp. gr., }60/60^\circ\text{ F.} = 141.5/(131.5 + \text{deg. A.P.I.}) \quad . \quad (499b)$$

For liquids lighter than water, the relation between the *Baumé degree* and the specific gravity is:

$$\text{Degree Baumé} = \frac{140}{\text{sp. gr. }60/60^\circ\text{ F.}} - 130 \quad . \quad . \quad . \quad (500a)$$

or

$$\text{Sp. gr., }60/60^\circ\text{ F.} = 140/(130 + \text{deg. Bé.}) \quad . \quad . \quad . \quad (500b)$$

The A.P.I. scale is approved by the Am. Pet. Inst. and the U. S. Bureaus of Standards and Mines for exclusive use in the U. S. petroleum industry, superseding the Baumé scale. However, much of the earlier information regarding oils was based on the older Baumé scale. For many purposes the two scales can be considered as being the same, the specific gravity for a Baumé reading being the same at 10 degrees and but 95/100 of 1 per cent less at 100 degrees than the gravity corresponding to the same reading on the A.P.I scale. A light oil has low specific gravity, but is high on the A.P.I. and Baumé scales, and vice versa.

American crude oils have specific gravities ranging from 0.82 to 0.97 (41 to 14° A.P.I.). Oil is sold by volume, the unit being the barrel of 42 U.S. gallons when it is marketed in bulk, and the gallon when small quantities are involved. The weight of a barrel or gallon of oil depends, of course, on the density of the fluid. The relations between the degrees A.P.I and specific gravity, and between them and the weights of a gallon of oil are shown graphically in Fig. 514.

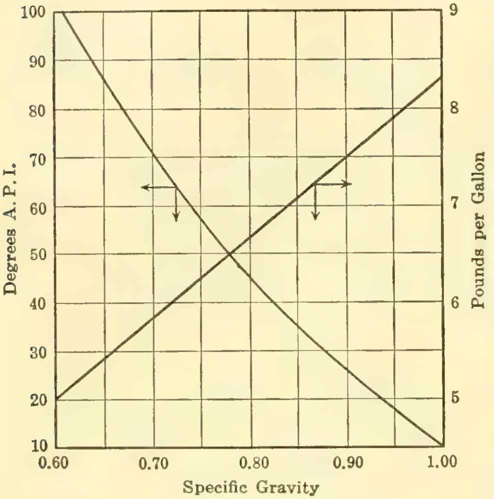

Fig. 514.—Relation between Degrees A.P.I., Specific Gravity, and Pounds per Gallon.

(f) The **higher heating** value per *pound* of crude petroleum varies quite regularly with the specific gravity of the material, and is expressed approximately by the following formula,[20] which may be assumed correct within 2 per cent,

$$\text{H.H.V.} = 18{,}650 + 40\,(\text{deg. Bé.} - 10), \text{ B.t.u. per lb.}, \quad (501)$$

Since the hydrometer reading on the Baumé scale increases as the density of the material becomes less, this equation indicates that the lighter oils have greater heat values than the heavier ones, when figured on a *weight* basis. The reverse is true for heat value per *gallon*, hence a barrel of light petroleum will have less heat value than a barrel of heavier oil.

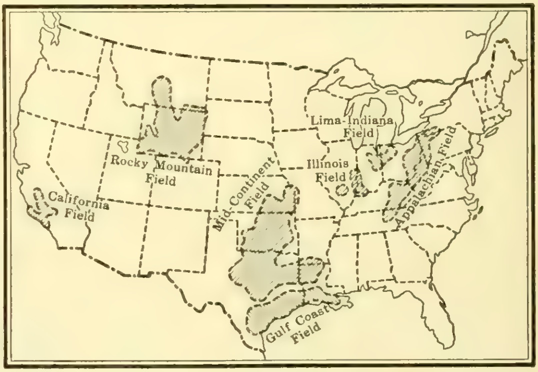

Fig. 515.—Map of Main Oil Fields in the United States.

(g) The locations of the **oil fields in the United States** are shown in a general way in Fig. 515. Petroleum is produced in almost half of the states of this country, and there is considerable importation from Mexico, Colombia, Venezuela, and Peru. This country has within its borders about 12 per cent of the world's resources in petroleum, and its annual production now is about 70 per cent of the total, or about 1,000,000,000 barrels per year. It has been very roughly estimated that this nation has already exhausted about one-half of its original inheritance of this source of energy. In the not distant future it will be necessary to depend largely on supplies of oil received from abroad or to use substitute oils derived from sources other than petroleum.

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[20] H. C. Sherman and A. H. Kropff, Jour. Am. Chem. Soc., Oct., 1908.

(**h**) The great variety of **hydrocarbon compounds**[21] of which crude oils are composed can be grouped in a number of regular series of the types given in Table XXXII. A single sample of crude oil will contain many hydrocarbons which may belong to the same or to several different series. The hydrocarbons in the Appalachian field consist almost wholly of paraffin compounds and are therefore said to be of the paraffin base. Proceeding westward through the United States the paraffins largely disappear and are replaced by the olefine and asphalt series. In the Texas and California fields the asphalts predominate, the oil being called of the asphalt base. The intermediate fields have mixed base oils. Mexican and South American oils have an asphalt base and Russian oils a naphthene base. But although crude oil has been subjected to examination for many years, its exact composition is

TABLE XXXII

Hydrocarbon Series	General Formula	First Member of Series
Paraffin or Methane	C_nH_{2n+2}	CH_4
Unsaturated* Olefines, or Saturated Naphthenes	C_nH_{2n}	C_2H_4
Acetylene, Unsaturated	C_nH_{2n-2}	C_2H_2
Asphalt	C_nH_{2n-4}	
Aromatics or Benzene	C_nH_{2n-6}	C_6H_6
Naphthaline	C_nH_{2n-12}	$C_{10}H_8$

* Unsaturated hydrocarbons are of such molecular arrangement that they can be made to combine with additional hydrogen, whereas it is impossible to combine additional hydrogen with saturated hydrocarbons.

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[21] For properties of several hundred hydrocarbons, see Day's "Handbook of the Petroleum Industry," published by John Wiley & Sons.

not known because of its extreme complexity and the difficulty of separating the various hydrocarbons in the laboratory.

360. The Petroleum Industry.[19]—(**a**) The petroleum industry deals primarily with production and refining, associated with which, however, are the very important operations of transportation and marketing. Oil from the wells is stored in steel tanks, some of 120,000 barrel capacity, or in concrete reservoirs which have been built of capacity as large as 3,000,000 barrels each. Transportation is by tank cars, ships (tankers), or pipe lines. The main trunk pipe lines of the country in 1930 are shown in Fig. 516, which omits district net works and gathering lines. Pipe lines range from 6 in. to 16 in. or more in diameter and modern ones are fitted for pressures of 900 pounds per square inch, with pumping stations from 25 to 70 miles apart, the requisite power being supplied by steam, oil, or gas engines, or by electric motors.

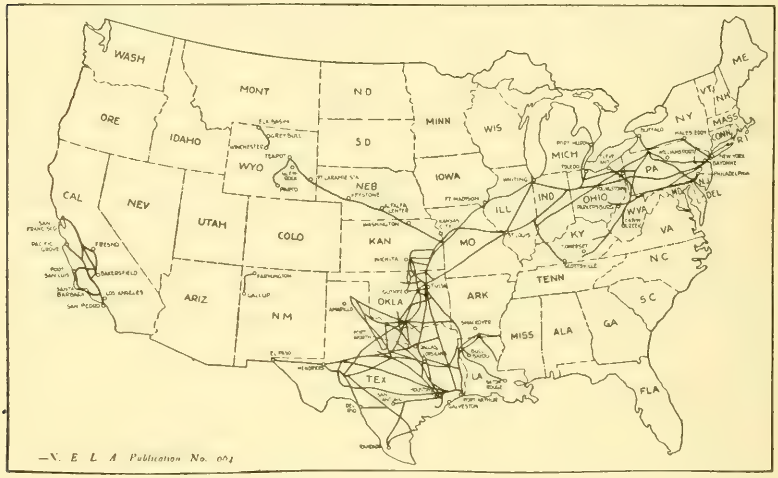

FIG. 516.—Oil Trunk Pipe Lines in the United States.

(**b**) In the **refining procedure** the crude petroleum is separated into its marketable components chiefly by the process of fractional distillation in which the fractions are boiled off successively, the vapors being condensed and collected separately to furnish (1) crude benzine, (2) kerosene distillate, (3) gas oil, (4) lubricating distillate, and (5) residuum. After being purified, the benzine is redistilled, producing a number of finished *gasolines* (varying from 54–74° Bé.) and a residue called *naphtha* (42–54° Bé.). The second distillate is purified to produce *kerosene* (about 42° Bé.), which may also contain the residue left after distilling the naphtha. The *gas oil* (32–40° Bé.), originally utilized mainly in manufacturing illuminating gas or for enriching coal gas, is now used in the manufacture of synthetic gasoline, especially when there is a deficiency of straight-run gasoline. *Fuel oil* is the residue after removing distillates (1) and (2), and usually (3), from the crude. *Lubricating oils*, obtained from the wax distillate, range from 24 to 32° Bé. From the residuum are obtained such products as waxes, dust-laying oils, petroleum asphalt (for road construction), and coke. The gasoline and kerosene distillates are refined to remove the sulphur, and the unsaturated hydrocarbons which have an offensive odor, and to provide the color desired.[22]

(**c**) The fractionation of crude is now accomplished very accurately in modern pipe stills equipped with bubble towers, and with heat balance carefully precalculated. Cracking processes are used to increase the gasoline output, from the original amount of from 15 to 30 per cent in the crude, to 50 per cent. While much larger percentages can be produced in this manner, it is not at present economical to do so. The equipment includes pumps to handle oil at from 350 to 700 deg. fahr. and pressures up to 2000 pounds per square inch, heaters, condensers, heat exchangers, and automatic control and recording apparatus. The fractioning-still heaters commonly discharge the product at 800 to 850 deg. fahr., and cracking stills discharge into the towers at 900 to 1000 degrees. The high temperatures and pressures require the use of special alloys, and thick walls, and careful design of parts.

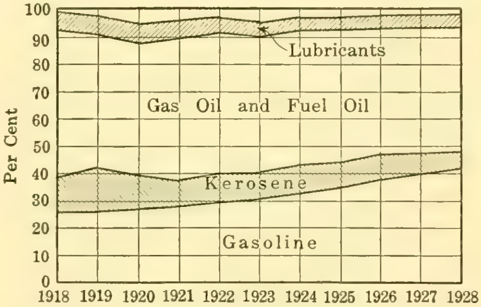

FIG. 517.—Percentage Yields of the Major Products of Petroleum.[23]

(**d**) The percentage yields of the major petroleum products in recent years are shown in Fig. 517.

The petroleum products in which the heat-power engineer is primarily interested are crude oil, gas and fuel oil, Diesel oil, gasoline, and kerosene.

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[22] For the distillation and other properties of various petroleums, see U. S. Bu. of Mines Bul. 291, "Tabulated Analyses of Representative Petroleums of the United States"; and Tech. Paper 346, "Properties of Typical Crude Oils from the Producing Fields of the Western Hemisphere," by Kraemer and Calkin.

[23] Bu. of Mines Bul. 318; Petroleum Refinery Statistics, 1928.

361. Thermal Properties of Petroleum Products.—Empirical equations have been derived by the U. S. Bureau of Standards to give some of the thermal and physical properties of petroleum products in terms of the specific gravity, temperature, and pressure of the fluid.[24] These equations are given in Table XXXIII.

The publication from which these equations were taken also contains tables from which the desired quantities can be read directly without computation.

This is the first attempt to supply the petroleum industry and users of petroleum products with comprehensive data on the thermal properties of these products in readily useful form somewhat analogous to the tables long available for steam and ammonia. It is fully realized, however, that the experimental data on which some of the properties are based are not as complete as could be desired, and that a revision may be necessary when additional information is obtained.

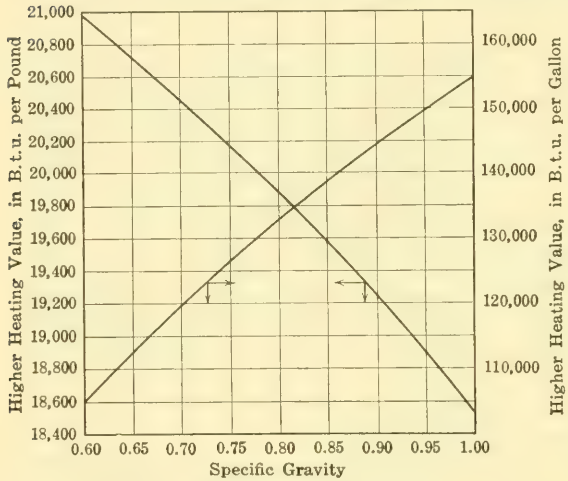

FIG. 518.—Relation between the Specific Gravity and the Heating Value per Pound and per Gallon of Petroleum Products.

The relations between the heating values per pound and per gallon and the specific gravity of petroleum products, as given by the Bureau of Standards, are shown by the curves in Fig. 518.

362. Fuel Oil.—(**a**) A fuel oil is defined in a report of the A.S.T.M.[25] as "any petroleum product used for generation of heat in a firebox. In general, any liquid or liquefiable petroleum product may be used as a fuel oil, the controlling factors being cost and adaptability. Fuel oils in common use generally fall into one of four classes: (1) crude petroleums, (2) fuel oils, (3) residual oils, (4) blended fuels." In addition to their use in furnaces, such oils are also used in Diesel and other oil

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[24] Misc. Pub. No. 97, "Thermal Properties of Petroleum Products." 1929

[25] Report of Subcommittee XX of Committee D-2, Nov. 2, 1923.

TABLE XXXIII

Empirical Equations for Thermal Properties of Petroleum Products

Property	Units	Empirical Equation	Equation No.	Experimental Range		Estimated Accuracy, Per Cent
				d	t	
Coefficient of expansion of liquid	$\frac{1}{V_{60}}\frac{dV}{dt}$	$=A+2B(t-60)$; $\log(A\times10^5)=0.835+0.70/d$, and $\log(B\times10^8)=(2.10/d)-1.20$	(502)	0.51–1.00	32–200	5
Heat of combustion at constant volume	B.t.u./lb.	$=22{,}320-3{,}780d^2$	(503)	0.51–0.99	....	1
	B.t.u./gal.	$=186{,}087d-31{,}515d^3$	(504)			
Specific volume of vapor	Ft.^{3}/lb.	$=\frac{0.242(t+460)}{p}\cdot\frac{(1.03-d)}{d}$	(505)	0.51–0.80	....	10
Thermal conductivity of liquid	$\frac{\text{B.t.u./in.}}{\text{Ft.}^2\text{/hr. °F.}}$	$=\frac{0.813}{d}[1-0.0003(t-32)]$	(506)	0.78–0.95	32–400	10
Specific heat of liquid	B.t.u./lb. °F.	$=\frac{1}{\sqrt{d}}(0.388+0.00045t)$	(507)	0.72–0.96	32–750	5
	B.t.u./gal. °F.	$=\sqrt{d}(3.235+0.00375t)$	(508)			
Latent heat of vaporization	B.t.u./lb.	$=\frac{1}{d}(110.9-0.09t)$	(509)	0.64–0.91	100–600	10
	B.t.u./gal.	$=925-0.75t$	(510)			
Heat content of liquid	B.t.u./gal.	$=\sqrt{d}(3.235t+0.001875t^2-105.5)=\mathrm{H}$	(511)	0.72–0.96	32–750	5
Heat content of vapor	B.t.u./gal.	$=\mathrm{H}+925-0.75t$	(512)	0.64–0.91	100–600	5

d = specific gravity at 60°/60° F.; t = temperature in °F; p = pressure in lb./in.^{2} absolute; unit volume of liquid = gallon, measured at 60° F.

engines. The term "fuel oil" is often applied to any oil, whether a petroleum product or not, that is safe to handle, has suitable physical properties, and is available in sufficient quantity at a low enough price, based on its heating value, to justify its use as a fuel. Usually fuel oil is crude oil from which gasoline, kerosene, and gas oil have been skimmed; but when there is an insufficient demand for the latter, it too is left in the fuel oil.

(**b**) The **heating value,** in B.t.u. per *pound* of fuel oil, ranges from 18,000 for heavy oils to 19,500 for light oils, but is usually about 18,500. It is given approximately by the equation,[20]

$$\text{H.H.V.} = 18{,}250 + 40(\text{deg. Bé.} - 10), \text{ B.t.u. per lb.,} \quad . \quad (513)$$

which may be assumed correct within 2 per cent. The specific gravity ranges from 0.85 to 1.0 (35 to 10 deg. Bé.). How the heating values per *pound* and per *gallon* vary with the density is shown in Fig. 518.

(**c**) The important properties of fuel oils to the engineer are: (1) specific gravity, (2) heating value, (3) flash and fire points, (4) pour point (cold test), (5) viscosity, (6) sulphur content, (7) moisture and sediment, (8) specific heat, and (9) coefficient of expansion. Equations for determining the thermal properties of fuel oils are given in Table XXXIII. For methods of testing these and other oils see Bureau of Mines Tech. Paper 323B entitled: U. S. Gov't. Master Specification for Lubricants and Liquid Fuels and Methods for Testing; also standard test methods are given in the publications of the Am. Soc. for Testing Materials.

(**d**) *Commercial* **standards** *for fuel oils* have been prepared by the U. S. Bureau of Standards[26] for six numbered grades, as follows: 1. light, 2. medium, and 3. heavy *domestic* fuel oils; 4. light, 5. medium, and 6. heavy *industrial* fuel oils for industrial or domestic oil burning equipment; all shall be hydrocarbon oils free from acid, grit and fibrous or other foreign matter likely to clog or injure the burner or valves; and the first four grades must also be free from water. The specifications give, in tabular form, the detailed requirements for each grade as to flash point, water and sediment, pour point, maximum viscosity, and distillation test. Specifications for Navy Standard, and Bunker Fuel Oils, Grades A, B, and C, for use by the U. S. Government and its agencies are contained in Bu. of Mines Tech. Paper 323B.

The *flash point* is the temperature to which the oil must be heated in order to give off sufficient vapor to form an inflammable mixture in air. It indicates the safety of the fuel on the one hand and the ease of ignition on the other. The flash points of grades 1 to 3, inclusive, range from 110 to 200 deg. fahr., and for the industrial grades the minimum is 150 deg. fahr.

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[26] Commercial Standards, CS12-29.

The *pour point* is the lowest temperature at which the oil will flow under prescribed conditions, and indicates the difficulty in handling or using the fuel at the lowest temperature to which it may normally be subjected. The maximum allowable pour point is specified as 15 deg. fahr.

The *viscosity* is the measure of resistance to flow through pipe lines, the cost of pumping, and the degree of atomization that may be secured with given equipment. As used in the specifications, it is measured by the seconds required for a definite volume of oil to pass through a small tube of specified dimensions at a specified temperature (using a Saybolt viscosimeter).[27] Viscosity decreases rapidly as the temperature increases, and preheating is necessary to make possible the use of oils of relatively high viscosity at normal temperatures. The maximum Saybolt viscosity of the last four oils ranges from 55 sec. at 100 deg. fahr. for No. 3 to 300 sec. at 122 deg. fahr. for No. 6 oil.

The *distillation test* of a sample under prescribed conditions gives an index of the volatility of the oil. The temperature at which 10 per cent of the sample is distilled over is called the 10 per cent point, and it indicates the ease of ignition of the oil. The 90 per cent point and end point temperatures are specified to make sure that the oil will burn completely and produce a minimum of carbon. The distillation test is required only for the domestic oils for which the 10 per cent point maximum ranges from 420 to 460 deg. fahr. The end point of No. 1 is 600 deg. fahr., and is not specified for the other grades. The 90 per cent point is 620 deg. fahr. for No. 2, and 675 degrees for No. 3.

(**e**) In 1928 about 47 per cent of crude oil was used as fuel oil. The amounts of fuel and gas oil, in million barrels, used in that year in the various fields was about as follows: steamships, 87; railroads, 71; export, 44; oil companies, 50; gas and electric power companies, 31; general manufacturing, 40; smelters, mines, iron and steel industry, 26; heating buildings, 22; navy, transports, and miscellaneous, 19.[28]

(**f**) **Diesel oil** is fuel oil that has the requisite fluidity to pass through the small passages met with in engines of this type, that is clean and non-corrosive, and that will not give carbon deposits; further, it should have the necessary qualities to lubricate the pump plungers. Suitable viscosity is of very great importance in such oil. Fuel oils of the U. S.

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[27] Viscosity is also determined by other methods, each giving a different result and known by a different name.

[28] "National Survey of Petroleum Oil Distribution," 1928. Report of U. S. Bu. of Mines.

Navy and bunker grades are not only furnace oils but are also classed as Diesel oils. At present the preparation of standard specifications for oils having the requisite characteristics for use in Diesel engines is being undertaken by a committee under the auspices of the A.S.M.E.

363. Kerosene.—This petroleum product is composed of a number of hydrocarbons of which $C_{12}H_{26}$ is about the average. Its specific gravity is from 0.78 to 0.82 (50 to 41° Bé.) and its higher heating value, which ranges from 20,100 to 19,800 B.t.u. per pound, may be found for its different densities from the Sherman and Kropff formula, as modified by Strong (Bu. Mines Bul. 43), which is

$$\text{H.H.V.} = 18{,}440 + 40(\text{deg. Bé.} - 10), \text{ B.t.u. per lb.} \quad . \quad (514)$$

The U. S. Govt. specification for ordinary kerosene, among other stipulations, require a flash point not less than 115 deg. fahr. and an end point of distillation of not more than 625 deg. fahr. The thermal properties of this fuel can be determined by means of the formulas in Table XXXIII, or the tables referred to in connection therewith.

This petroleum fraction, once the main product from crude oil because of wide use in kerosene lamps and for fuel, is now a minor product and is applied but little in the generation of power.

364. Gasolines. — (**a**) The hydrocarbons of the paraffin series (C_nH_{2n+2}), which form the chief constituents of eastern petroleums, are generally made the basis for theoretical discussions of gasolines. The gaseous and liquid members of this group are given in Table XXXIV.[29]

Starting with methane or marsh gas (CH_4), at ordinary temperatures the first four members of the series are gaseous, the next twelve are liquids, and the remainder solids. Octane (C_8H_{18}) is about the average of the mixture of eight (more or less) hydrocarbons contained in a representative heavy-grade gasoline. Before the introduction of the automobile, gasoline had lighter components than now, and was a cheap by-product for which there was little use. With increasing demand and with improvements in carburetors, more of the slightly heavier hydrocarbons have been included, the final temperature of distillation being gradually raised from 250 deg. to about 440 deg. fahr., with the result that the recovery of "straight refinery gasoline" from crude oil has been increased in some cases from 10 per cent to 25 per cent or more. By using the *cracking* processes (such as those of Burton, Hall or Rittman), in which the transformation occurs under high pressures which change the vaporization temperatures of the different fractions and

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[29] Heldt's "The Gasoline Automobile," Vol. IV (Fuels and Carburetors) Also see Marks' "The Airplane Engine."

TABLE XXXIV

General Properties of the Paraffin Series (Heldt)

Name	Chemical Formula	Molecular Weight*	Boiling Point, Deg. Fahr.	Specific Gravity at 60° Fahr.	Per Cent of Carbon
Methane	CH_4	16	−260	Gas	75.00
Ethane	C_2H_6	30	−140	Gas	80.00
Propane	C_3H_8	44	−46	Gas	81.85
Butane	C_4H_{10}	58	34	Gas	82.76
Pentane	C_5H_{12}	72	98	0.628	83.33
Hexane	C_6H_{14}	86	157	0.672	83.73
Heptane	C_7H_{16}	100	209	0.697	84.00
Octane	C_8H_{18}	114	256	0.713	84.21
Nonane	C_9H_{20}	128	300	0.728	84.37
Decane	$C_{10}H_{22}$	142	338	0.741	84.50
Undecane	$C_{11}H_{24}$	156	374	0.754	84.61
Dodecane	$C_{12}H_{26}$	170	410	0.767	84.70
Tredecane	$C_{13}H_{28}$	184	443	0.780	84.78
Tetradecane	$C_{14}H_{30}$	198	476	0.793	84.84
Pentadecane	$C_{15}H_{32}$	212	510	0.807	84.90
Hexadecane	$C_{16}H_{34}$	226	540	0.820	84.95

* Strictly, this column should now be called the "Formula Weight" as explained in Sect. **371** (c).

cause the heavier compounds to break up and form those of higher volatility, 63 per cent or more of the crude oil can be converted into motor fuel. Gasoline is also recovered from *natural gas* by either the compression process or by the absorption process. The name *casing-head gasoline* is applied to that obtained from the gas from oil wells. Although too volatile for use alone in internal combustion engines, such gasoline is adapted for blending with the heavier naphthas, or with other volatile products, such as those obtained from cracking processes, to form *blended gasoline*; and it is a component that facilitates starting in

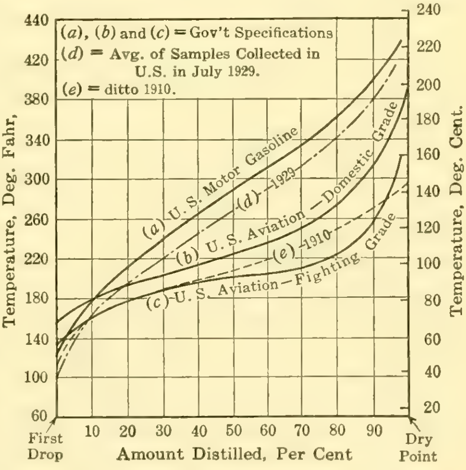

Fig. 519.—Gasoline Distillation Curves.

cold weather. The "high test" or aviation gasoline has less of the heavy ends of distillation and hence is more expensive than "motor fuel" of the ordinary U. S. specification. Fig. 519 gives the distillation curves for these and other gasolines, including the average of the semi-annual survey as of July, 1929, of motor gasoline sold throughout the country.[30] The Federal specifications for gasolines and methods of testing this fuel are given in U. S. Bu. of Mines Tech. Paper 323B.

(**b**) The **thermal properties** of gasoline are represented by the equations given in Table XXXIII or are given in the Tables included in the U. S. Bureau of Standards Miscellaneous Publication No. 97. The specific heats, latent heats, densities, pressure-temperature relations, dew point, etc., of any given sample of gasoline depend on the kinds and proportions of the hydrocarbons composing it. These factors, together with the laws of flow of air and vapors, and the law of partial pressures, enter into the design and operation of carburetors.

(**c**) The **higher heating value** of gasoline, as found by calorimeter tests, agrees quite closely with the quantities estimated by the Sherman and Kropff formula which, as modified by Strong (U. S. Bu. Mines Bul. 43, p. 19), is

$$\text{H.H.V.} = 18{,}320 + 40\ (\text{deg. Bé.} - 10),\ \text{B.t.u. per lb.} \quad . \quad (515)$$

The average H.H.V. of commercial gasoline is 20,200 B.t.u. per *pound* with only 1.5 per cent difference over a range of specific gravity from 0.687 to 0.745 (74 to 58 deg. A.P.I.); but per *gallon* there are 116,500 B.t.u. for specific gravity 0.687, and 124,000 B.t.u. for specific gravity 0.745, the heavier fuel yielding over 7 per cent more heat per gallon than the lighter one. The flash point of gasoline is generally below 70 deg. fahr. For the combustion of 1 pound of gasoline there is needed about 15 pounds of air, which has a volume of about 200 cubic feet at 62 deg. fahr. and one atmosphere.

(**d**) The calorific value alone, however, is not a true measure of the usefulness of a gasoline or motor fuel; the fuel must also have sufficient volatility to permit an evenly distributed explosive mixture to be delivered equally to the cylinders not only for the normal operation of the motor, but also for cold starting; and further, the fuel should have *anti-knocking* qualities that will permit the use of the high ratio of compression that is necessary for obtaining high efficiencies.

(**e**) The **volatility** of the gasoline is indicated by its distillation curve, especially by the upper portion; but it must be remembered that to vaporize the higher fractions it is not necessary to use the corresponding distillation temperatures shown on the curve (Fig. 519). Dalton's law applies the same as for vapors of water and other liquids. Gasoline will evaporate into the atmosphere even though the temperature is near freezing, provided the air is not already saturated with this vapor at the partial pressure that corresponds to the existing temperature. In a combustible mixture the gasoline vapor occupies only a small proportion of the space filled by the mixture and the partial pressure of the gasified fuel is low. *Volatility* may be defined as the degree to which a fuel will vaporize under certain equilibrium conditions which must be specified.

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[30] Serial No. 2959 of U. S. Bureau of Mines.

(f) The **distillation curve** shows important characteristics of the fuel. Apparently the 10 per cent point of the curve is the criterion of the ease in starting a gasoline engine [31] as well as of the vapor lock (or interrupted flow due to boiling in the fuel feed system), the middle part has an important bearing on the rapidity of acceleration and ability to keep the engine running, and the 90 per cent point [32] of the curve indicates the dew point at which liquid fuel will deposit on the cylinder walls, causing dilution of crankcase oil, and also indicates the temperature to which the mixture must be heated before it becomes dry. The end point is probably the least important point on the curve.[33]

(g) The **detonating characteristics** of a gasoline limit the amount of compression that may be used in an engine. The paraffin hydrocarbons have poor anti-knocking qualities under high compression. In this series, octane (C_8H_{18}) is comparatively free from knocking while heptane (C_7H_{16}) is a violent knocker and practically all commercial gasolines fall between these two as to detonation qualities. Using these two hydrocarbons as standards, the **anti-knock value** of a sample of gasoline may be specified by the number of parts by volume of octane that must be added to 10 parts of normal heptane to give the same knocking characteristics as the sample under investigation.[34]

(h) A gasoline having unsatisfactory detonation characteristics may be improved in this respect by blending with it some low-knock fuel such as benzol (C_6H_6), toluene (C_7H_8), or other aromatics or naphthenes, alcohol, etc., which stand high compression without "pinking"; or it may be treated with a very small quantity of tetra-ethyl lead (to make "ethyl" gasoline), or with other anti-knock compound or "dope" that will decrease the rate of flame propagation.[35]

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[31] See Soc. Automotive Engrs. Journal, March, 1927, p. 353 and Oct., 1929, p. 345.

[32] S. A. E. Journal, Nov., 1928, p. 478.

[33] The "Relation of Motor Fuel Characteristics to Engine Performance," by G. G. Brown, U. of Mich., Eng'g. Research Bul. No. 7.

[34] Proposed Standard, S. A. E. Journal, Aug., 1930, p. 241.

[35] "Detonation," by W. A. Whatnough, The Automobile Engineer, beginning Feb., 1927; "Dopes and Detonation," by H. L. Callendar, Engineering, beginning Feb. 4, 1927, or S. A. E. Journal, March, 1927. Also recent articles in the S. A. E. Journal.

365. Methods of Increasing the Gasoline Output.—(a) In producing gasoline by the usual distillation processes only 15 to 30 per cent of the crude oil is converted into this form of fuel for which there is such a tremendous demand; and while much of the remainder of the crude can also be made into useful products by fractionation, yet there is often great lack of economic balance between the outputs, or possible productions of such products, and the demand for them and for gasoline. More gasoline can, of course, be produced by such processes by using more crude petroleum, but this method involves great waste. It is desirable therefore to have means for balancing the output of gasoline and other products obtainable from the crude so as to accord with the demands, thus conserving the unreplenishable natural petroleum resources which are being so rapidly exhausted, and lowering the prices of all the products. Such a balance can be effected by employing the *cracking* and the *hydrogenation processes.*

(b) Essentially **cracking** consists of heating oils to such a temperature that their heavy fractions decompose into lighter ones with the production of mixtures of lower boiling point. The function of the increased temperature is to break the bonds of the groups that make up the more complex hydrocarbons. In many processes the high pressure is of chief importance in controlling the temperature of distillation, but it also exerts an influence on the nature of the decompositions produced. There are a great many different processes by which the cracking can be accomplished, but, in most cases, the cracking is done at pressures above 600 lb. per square inch and temperatures above 900 deg. fahr. The product differs chemically from straight-run gasolines in containing considerable quantities of unsaturated and aromatic hydrocarbons. At present about 30 per cent of the gasoline of this country is produced by this method.

(c) In the **hydrogenation process** as applied to oils the basic principles of chemical synthesis are used to build up new hydrocarbons by the addition of hydrogen to the old ones, whereas the cracking processes are destructive methods and not all of the heavier oils can be converted into lighter ones. With the recently developed methods of hydrogenation, it is possible, by introducing hydrogen under pressures of from 2800 to 4200 lb. per square inch and with the assistance of a catalyst, to build up the hydrocarbons desired and to convert approximately all of the original oil into gasoline or other products. This process is now being developed on an extensive commercial scale by many of the oil companies. By it the gasoline production from crude oil can be made about twice its former amount, still leaving sufficient margin for the production of other petroleum products of high economic value and in proportions in which they appear to be required.

366. Sources of Petroleum Substitutes.—(a) Although nature's supply of petroleum will probably not last many generations, the oil situation will not become serious because, when economic conditions justify it, substitute oils can be depended upon. The deficiency of this natural product can be made up by liquid fuels abstracted from oil shale and coal, and also from vegetable matter which is constantly being formed.

(b) **Shale Oil.**—There are enormous deposits of sedimentary shale rocks containing fragments of organic matter, such as fossil plants and animals, which can be distilled to yield oil which can be refined in a manner similar to that used with crude petroleum, and much the same fractions can be obtained. With the known methods of producing shale oil, however, the product is too costly at present in most localities to compete with crude petroleum.[36]

(c) **Carbonization of Coal.**—The destructive distillation of coal, in the absence of air, has been used for many years to convert coal into coke, coal gas, tar, and other valuable products useful in the chemical and other industries. So far, the distillation of coal has been carried out mainly by the **high-temperature carbonization process,** which carries the coked charge to temperatures of from 1800 to 2100 deg. fahr. Recently much attention has been given to the development of **low-temperature carbonization processes** to produce smokeless solid fuels from high-volatile coals, together with liquid and gaseous fuels, the maximum temperature of the charge being from 1000 to 1400 deg. fahr. The methods employed generally use externally heated retorts in which the charge of coal is stirred or in which the coal is arranged in thin stationary layers, or they use internally heated retorts filled with coal that is heated by hot gases or superheated steam or in which coal in the pulverized form is heated in suspension by means of hot gases. The method adopted should depend mainly on the characteristics of the coal and on the products which are to be made. These products differ materially from those obtained in the high-temperature process. The yield of liquid fuel by the carbonization processes is from 20 to 40 gallons per ton of coal; and while such oils would not replace petroleum products to any very large extent, nevertheless their production is even now of economic value in countries having coal but no petroleum resources of their own. With low temperatures the yield of oil by carbonization is greater than with high temperatures.

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[36] Reference: M. J. Garvin, "Oil Shale," U. S. Bu. of Mines Bul. 210. This also contains an excellent bibliography on the subject.

(d) **Coal Liquefaction by Hydrogenation.**—In coal the C/H ratio averages about 16, whereas in petroleum it is about 8. Therefore, by combining more hydrogen with the carbon in coal it should be possible to approach the chemical composition of the oil. By the Bergius method of hydrogenation of coal, for example, about 50 per cent of the solid fuel is converted into oils of various grades. In this process oil or tar is mixed with powdered coal to form a paste which is pumped under a pressure of 1500 to 2200 pounds per square inch into a steel tube into which hydrogen and iron oxide are also introduced, the latter to act as a catalyst. The mixture is then heated to 750 to 800 deg. fahr. and yields about 15 per cent of motor spirit, 15 per cent of Diesel and creosote oil, 20 per cent fuel oil, lubricating oil, and pitch, and 20 per cent gas.[37]

(e) **Synthetic Methods.**—Liquid fuels can also be produced from coal, peat, oil shale, waxes, etc., by synthetic processes. For example, in the method used by F. Fischer,[37] the raw material is first converted to carbon monoxide (CO) and this is then combined with hydrogen under low pressure and a temperature of from 400 to 575 deg. fahr., with cobalt and iron as catalysts, to produce the liquid and solid compounds ordinarily derived from petroleum. In other processes the gases from the coal are converted by pressure synthesis into methanol, synthol, and other liquid combustibles.

367. Alcohol.—(a) Both *methyl* ("wood") and *ethyl* ("grain") alcohol are used as fuel to a limited extent. Methyl alcohol (CH_4O), which is poisonous, was formerly made solely by the dry distillation of wood, but now it is principally a synthetic product formed by passing highly compressed carbon dioxide and hydrogen over a catalyst. Ethyl alcohol (C_2H_6O) is made by a fermentation and distillation process from grain, fruit, or vegetable matter containing starch or sugar, or by the treatment of wood with sulphuric acid. *Synthetic alcohol* can be produced from water gas, coal, and other sources.

(b) Methyl alcohol, or methanol, has higher and lower heating values, respectively, of 9600 and 8410 B.t.u. per pound. The higher calorific value of *absolute* ethyl alcohol (i.e., containing no water) is about 12,820 B.t.u. per pound and the lower value is about 11,580 B.t.u. per pound. The heat value of "Commercial" ethyl alcohol, containing about 10 per cent of water, by volume, varies with the materials used in denaturizing; the higher and lower values are generally about 11,600 and 10,500 B.t.u. per pound, respectively.

(c) Alcohol has many advantages and some disadvantages over gasoline as a fuel for certain purposes, but at present its relative cost, in this country at least, is so great as to prevent its extensive use as such. Adding alcohol to gasoline increases the volatility of the fuel, and in an internal combustion engine using the mixture the compression ratio can be increased without causing knocking, thereby improving the thermal efficiency and power obtainable from a given piston displacement. However, if combustion is not complete, acetic acid will form and cause corrosion, to prevent which benzol may be added to the mixture.

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[37] Proceedings of International Bituminous Coal Conference at Pittsburgh, 1926.

368. Gaseous Fuels.—(a) **Natural gas** is available in enormous quantities in the United States, the main fields being about the same as those shown for oil in Fig. 515. The natural gas resources in certain regions are so great that it has become economically feasible to pipe the gas to far distant centers of consumption. The trunk pipe lines for natural gas in the United States (1930) are shown in Fig. 520. Many of the lines are from 8 to 22 inches in diameter, with compressor stations operating at 400 lb. per square inch and located from 50 to 100 miles apart. The most recent line (1930) is that shown dotted in the figure. It is 1000 miles long, 24 inches in diameter, transmits the gas at pressures of from 300 to 600 lb. per square inch and is expected to require a total of about 100,000 horsepower to operate the compressors in the various stations along the line. Natural gas has about double the heating value, per cubic foot, of city gas, and much of it is used in the more distant places to enrich the manufactured product to bring it to the legally prescribed values.

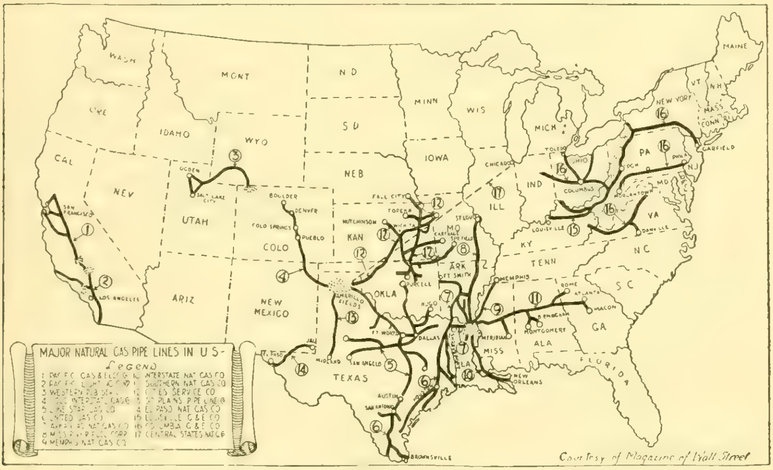

FIG. 520.—Natural Gas Trunk Pipe Lines in the United States.

Natural gas is a mixture of combustible and incombustible gases (the latter generally occurring but in very small quantities); and the constituents and their proportions seldom are the same in different districts, or in the different strata in the same district. The components are mainly hydrocarbons, particularly methane (CH_4), and often include considerable amounts of ethane (C_2H_6) and hydrogen, small percentages of CO, CO_2, O_2, and N_2, and traces of helium. Representative analyses of four gases, from different important districts, are given in Table XXXV.[38]

TABLE XXXV

REPRESENTATIVE ANALYSES OF NATURAL GAS FUELS

Source	Per Cent by Volume			
	A	B	C	D
	Ohio	Texas	South. Cal. Edison Co., Long Beach	Signal Hill, Calif.
Carbon Dioxide (CO_2) . . .	0.22	0.2	0.1	0.9
Unsaturants (C_n H_{2n})	0	0	0	0
Hydrogen (H_2)	1.82	0	0	3.6
Carbon monoxide (CO) . .	0.45	0	0	0
Methane (CH_4)	93.33	94.4	74.7	62.5
Ethane (C_2H_6)	0.25	2.9	23.7	32.9
Oxygen (O_2)	0.35	0.2	0.1	0.1
Nitrogen (N_2)	3.40	2.3	1.4	0
Hydrogen sulphide (H_2S) .	0.18	0	0	0
Total	100.00	100.0	100.0	100.0
H.H.V., B.t.u. per cu. ft. . .	1,015	1,000	1,221	1,211
Specific gravity (Air = 1) . .	0.570	. . . .	0.701	. . . .

Most natural gas is first stripped of its readily condensible hydrocarbons, which are then blended with motor fuels or used for other purposes. The higher heating values of this gas range from about 700 to 1400 B.t.u. per cubic foot,[39] under standard condition. In the A.S.M.E. Test Code the standard condition is taken at 68 deg. fahr. and 29.92 in. Hg, but gas manufacturers frequently take the temperature as 62 deg. fahr.

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[38] From N.E.L.A. Prime Movers Com. Report on "Burning of Liquid and Gaseous Fuels," June, 1930.

[39] Bu. of Mines Tech. Papers 10, 104, and 109 are devoted to Natural Gas.

(b) **Blast furnace gas,** a by-product of the iron industry amounting usually to about 70 cubic feet of gas per pound of coke used in the furnaces, is useful as a fuel in gas engines and in furnaces but first must be cleaned (see Fig. 12, Part I). Its analysis on the volumetric percentage basis is about as follows: CO, 28; CO_2, 11; H_2, 2.8; CH_4, 0.2; and N_2, 58. Its higher heating value is from 85 to 110 B.t.u. per cubic foot under the standard condition. The lower calorific value is about 2 B.t.u. per cubic foot less than the higher value.

(c) **Producer gas**[40] made by the incomplete combustion of coal by the process briefly described in Sect. **374** (b), may be used in internal combustion engines or in furnaces. It can be made from wood, peat, lignite, bituminous or anthracite coal, or from oil, and may be the means of utilizing inexpensive lower-grade fuels which are otherwise useless. Its combustible part contains a large proportion of carbon monoxide (CO). If the CO formed is allowed to cool before it is burned there will be a heat loss of 28 per cent, as will be explained in Sect. **372** (i). A similar loss results when producer gas is allowed to cool before being used as a fuel. However, if the *hot* gas is immediately delivered, without heat loss, to a furnace and is completely burned therein, the efficiency would be 100 per cent under ideal conditions. Therefore, when circumstances permit, the gas should be used directly as it comes from the producer, thereby avoiding this loss. When the gas is cooled due to being conducted to great distances, or being washed, or being stored, the efficiency is unavoidably reduced. It has been suggested that producers can be placed directly under boilers or can be used in direct combination with industrial furnaces, and that high efficiencies can thus be obtained even when using a fuel of poor grade, possibly one otherwise worthless, and consequently relatively cheap.

The analysis of producer gas varies widely, the volumetric percentages of the constituents being about as follows: CO, from 14 to 27; H_2, 0 to 18; CH_4, 0.2 to 3.5; CO_2, 5 to 12; N_2, 45 to 60. The higher heating value ranges from 110 to 175 B.t.u. per cubic foot, standard condition.

(d) **By-product coke-oven gas,** together with tar, ammonia, and other valuable products are obtained in the manufacture of coke by the destructive distillation of coal. The gas from this high-temperature carbonization process has a higher heating value varying from 400 to 600 B.t.u. per cubic foot (std. cond.). Its main constituents are CH_4 and H_2 in varying proportions. **Gas from low-temperature**

[40] "Fuel Saving in Modern Gas Producers and Industrial Furnaces," Trans. A.S.M.E., 1921, and "Progress in Gas Producer Practice," Trans. A.S.M.E., 1928, both papers by W. B. Chapman; also Bu. of Mines Bul. No. 7.

carbonization processes ranges in calorific value from 700 to 1000 B.t.u. per cubic foot.[41] Under favorable conditions these two gases can be used economically to generate power.

Illuminating gas (coal gas, manufactured gas, or city gas), formed by distillation of bituminous coal, is ordinarily too expensive to use for the generation of power; and the same is also true of blue and carburetted **water gases,** which are made by a number of reactions which are given in Sect. **374** (c).

In the formation of *blue water gas*, the process is intermittent: first, air is passed through the fuel bed to produce the necessary heat for the process by burning some of the carbon of the fuel to carbon monoxide (CO); and then, steam is substituted, with the formation of CO and hydrogen, until the fuel bed becomes cooled to a point requiring the introduction of air again, the alternations being repeated continuously. The resulting gas composition therefore varies during the process. The gas burns with a blue flame, as it contains practically no illuminants (unsaturated hydrocarbons) and is valuable as a heating gas. The volumetric percentages of the constituents of this gas are about as follows: CO, 41; H, 49; CO_2, 4.2; N, 4.3 with traces of O and CH_4. The H.H.V. is about 307 B.t.u. per cubic foot (std. cond.)

Carburetted water gas is blue gas with about 5.9 per cent of illuminants added from cracked gas oil, the H.H.V. being about 483 B.t.u. per cubic foot.

(e) **Gas is an ideal fuel;** its smokeless combustion, requirement of but little excess air, cleanliness, controllability and convenience are unapproached by any other type of fuel. When the exhaustion of our anthracite and petroleum resources have progressed sufficiently far, economic conditions may bring about the use of various combination processes for producing from cheap volatile coal a smokeless fuel for domestic use, fuel-oil, and gasoline substitutes, a large yield of cheap gas and other valuable constituents, devoting each to the highest use for which it is adapted. It has been suggested by Dr. A. S. Fieldner[42] that the gas works of the future could equalize their load by making gas in winter, and, when the weather gets warm, using their excess gas manufacturing capacity for gasoline production. Since the gasoline can be stored, the gas plant can be operated at maximum capacity throughout the year, thus reducing the excessive capital cost due to the maximum demand capacity required under our present system of gas manufacture. Under these conditions manufactured gas may become relatively cheap and have much wider application than it now has.

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[41] Bu. of Mines Tech. Paper 396, "Low-temperature Carbonization of Coal," by A. C. Fieldner.

[42] Lecture at Johns Hopkins Univ., Apr., 1925.

369. Future Developments.—(a) Various possibilities of meeting the changed fuel situation which will develop as the natural resources in petroleum, gas, and the better grades of coal become depleted and when the economic conditions are altered, have already been suggested in the foregoing discussion. However, there are many other possibilities. The arts of producing, preparing or manufacturing, and utilizing fuels of the various kinds and their products will probably undergo many changes in the future, and an enormous amount of investigation is necessary, not only of such matters, but also of the economic factors involved. Even now many extensive researches are being conducted, and reports of the developments taking place appear so rapidly that important references of today may become obsolete in a short time. Hence, one interested in this subject should consult the most recent articles, reports, bulletins, etc., as well as the former literature on it.

(b) To use a fuel to the greatest advantage one must know the basic principles of combustion and be acquainted with the different kinds of combustion apparatus and methods. Some of the succeeding chapters will be devoted to the consideration of these matters.

ADDITIONAL REFERENCES

"Classification of Coal and Specifications," Preliminary Report of A.S.T.M.; also in Power Plant Engineering, July 1934, p. 334.

"Fuels and Their Combustion," by R. T. Haslam and R. P. Russell. McGraw-Hill Book Co., New York.

"Gas Engineers' Handbook," by F. Wills and others. McGraw-Hill Book Co., New York.

"Handbook of Oil Burning," by H. F. Tapp. Am. Oil Burner Assoc., New York.

"Principles of Motor Fuel Preparation and Application," Vol. I, by A. W. Nash and D. A. Howes. John Wiley and Sons, New York.

"Progress in Fuels" (with Bibliography), Trans. A.S.M.E., 1933, FSP-55-1, p. 1.

"Standards for Gas Service," U. S. Bureau of Stds. Circ. 405, 1934.

"International Coal Carbonization," by John Roberts and Adolf Jenkner. Pitman and Sons, London, 1934.

CHAPTER XXVIII

GENERAL PRINCIPLES OF COMBUSTION

370. Combustion—(**a**) Some chemical reactions are accompanied by the absorption of heat and others by its liberation. The first are **endothermic reactions** and the second **exothermic ones.** During these combinations, the amount of heat energy liberated or absorbed, termed the **heat of reaction,** is (1) independent of the time occupied in the reaction, (2) proportional directly to the weight of the material taking part in the process, (3) dependent slightly on whether that process is carried out at constant pressure or at constant volume, and (4) is affected both by the initial temperatures of the materials reacting and by the final temperature of the products formed. In Combustion Engineering the heat of reaction is also commonly referred to as the *calorific value*, the *heat of combustion*, the *heating value*, or the *heat value* of the fuel.[1]

(**b**) The **object of combustion,** in heat-power engineering, is either (1) the orderly, direct, and rapid production of heat which is to be transferred to other bodies, as in the furnace of a boiler, (2) the development of mechanical energy by means of the direct action of the products of combustion on a piston, as in the cylinder of an internal combustion engine, or (3) the formation of a more suitable kind of combustible—such as an artificial gas or coke—from the original material.

The usual *combustibles* are composed almost wholly of carbon (C), hydrogen (H), and hydrocarbons (C_nH_m), but often contain small percentages of sulphur (S). They are burned by combining them with oxygen (O).

(**c**) *Combustion,* to the engineer, means the chemical reaction between *oxygen* and *fuels,* that results in the evolution of heat at such a rate as to cause an increase of considerable magnitude in the temperature of the substances involved. The oxygen is supplied from the air, which, however, is composed chiefly of nitrogen (N). If the heat evolved by a reaction is generated at such a gradual rate that it is dissipated without causing sensible temperature rise the process constitutes **slow combustion,** and is useless for the production of heat or power. To initiate **rapid combustion,** most substances require that their temperature be raised above ordinary values by means of heat supplied from external sources.[2] The temperature at which rapid combustion becomes independent of external supplies of heat is known as the **ignition** or **kindling temperature.** Starting with slow combustion, if the rate of heat-dissipation is diminished there may result a rise in temperature of sufficient magnitude to cause **spontaneous ignition** which is followed by rapid combustion, such as sometimes occurs in coal heaps or in piles of oily rags which are left in confined places. Some substances will cease to burn unless the application of heat is continued. Some reactions occur with such a rapid increase in temperature and velocity that they culminate in an **explosion** or **detonation.** To be useful as a combustible for ordinary purposes the material must be such that, after ignition has once occurred, the process will be self-sustaining and the chemical reaction will proceed until equilibrium has been established, and do so without detonation.

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[1] Note that these terms are not to be confused with the *heat content* or the *total heat.*

(**d**) The **quickness of ignition** and **rate of combustion** of *solid and liquid fuels* depend principally on the following factors: (1) The *temperature* available—e.g., when the temperature is near 60 deg. fahr., the rate of most reactions is doubled for each 18 deg. fahr. rise; and around 1800 deg. fahr. it requires an increase of about 180 degrees to produce the same result. (2) The *rapidity* with which the air and fuel are brought together. (3) The character of the *surface* of the fuel. (4) The *ratio of surface area to volume* of the substance—this ratio being very much greater when the solid fuel is powdered and when the liquid fuel is finely atomized than when such fuels are burned in large masses. (5) The rate of *diffusion* of air to the fuel surface. A motionless gas film adjacent to a solid or liquid resists diffusion, but the thickness of this film may be decreased by the *scouring action* resulting from passing the air over the surface at high velocity.

With *gases*, the rapidity of ignition and burning are dependent primarily on (*a*) *temperature,* (*b*) intensity of *pressure,* (*c*) relative quantities (*concentrations*) of gas and oxygen (air), and (*d*) the *catalyzing action of hot surfaces* with which they are in contact—e.g., incandescent firebrick walls; also (*e*) certain materials such as platinum, nickel, ferric oxide, etc., may act as *specific catalysts,* particularly at moderate temperatures, and (*f*) in some cases the presence of *traces of water* will hasten the reactions.[3]

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[2] Although most substances require this rise in temperature before combustion can ensue, some chemicals do react immediately and spontaneously when brought together, but such cases are not encountered in ordinary combustion engineering.

[3] "Fuels and Their Combustion," Haslam and Russell.

(e) **To insure complete combustion** of all parts of the fuel present, in whatever form, the following conditions must be fulfilled: (1) The fuel and air must be properly ***m**ixed* at all points in the combustion zone, (2) sufficient ***a**ir* must be supplied to furnish the required oxygen, (3) the ***t**emperature* at those points must be adequate to maintain combustion, (4) there must be enough ***t**ime* allowed for the process to be completed —e.g., the volume and arrangement of the combustion space in the furnace of a boiler or in an industrial furnace should be such that the flame will completely burn out before encountering the dampening effect of relatively cold walls such as the boiler tubes. These four requirements are referred to frequently in connection with combustion engineering and they can be recalled by means of the key word **matt,** formed from the initial letters indicated in heavy type.[4] To insure that oxygen (air) in adequate quantity will be in contact with every part of the body of fuel, even though the distribution is not uniform, it is usual to supply an amount somewhat in excess of that just sufficient for complete combustion. The latter amount is commonly called the "*theoretical air*," and the additional amount is "*excess air.*"

(f) When the ultimate chemical analysis of the fuel is known, the simple chemical reaction equations can be applied for determining the kinds and amounts of products that should result and also the quantity of heat that should be made available. And the analysis of the actual *exhaust gases* issuing from an internal combustion engine, or the *flue gases* coming from a boiler or industrial furnace, can be compared with the ideal composition of such gases, and thereby much valuable information can be derived regarding the fuel and the character of the combustion process. Hereafter these gases will be called collectively the *exit gases*, and in the discussion which follows it should be remembered always that, unless otherwise stated, the statements apply to the combustion processes in both of these types of apparatus. But before proceeding further it is necessary to review briefly the chemical reactions involved and to have available certain combustion data.

371. Some General Principles Regarding Chemical Reactions.—(a) The *atomic weight,* or more strictly the *combining weight,* of any chemical element is its weight relative to that of the oxygen atom, for which the value of 16 has been arbitrarily agreed upon. With this sytem, the atomic weights of the important elements involved in combustion are as follows: carbon, 12.000; hydrogen, 1.008; sulphur, 32.065; nitrogen, 14.008. The whole numbers are sufficiently accurate for most of the calculations involved in combustion.

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[4] "Combustion in the Power Plant," T. A. Marsh.

(**b**) The *molecule* of a *gas*[5] is now considered the smallest part that has the physical and chemical properties of the substance. Whether solids and liquids are made up of molecules is a question that involves a precise definition of what a molecule of a solid or liquid really is. Lack of agreement regarding such a definition often causes the statements of different scientists to be somewhat inconsistent in this matter. Concepts regarding the molecular construction of matter have undergone radical changes in recent years as a result of the application of X-rays to the study of crystal structure. Nearly all solids are now known to be crystalline.[6]

The basic ideas regarding the structure of solids and liquids are important to engineers in the study of combustion because many of the most valuable fuels are in these forms, and the use of the term *molecular weight* in such cases may be confusing and unjustified. For example, carbon in coal constitutes our chief fuel, yet its molecular weight is unknown. The atomic weight of carbon has long been known as 12, but the molecular weight of this element has not been definitely determined because of the very great difficulty of securing carbon in the form of vapor or gas at any reasonable temperature. Therefore, in very recent years the tendency is to consider the term *molecular weight* strictly applicable to *gases* only.

(**c**) The term **formula weight,** on the other hand, may be applied to any solid, liquid, or gas for which the *chemical formula shows the kinds and the comparative numbers of atoms in the substance,* regardless of what may, or may not, be known as to its molecular structure and, therefore, its molecular weight. For example, the formula weight of CO is $12 + 16$ or 28; of CO_2, $12 + 2 \times 16$ or 44; and of H_2O, $2 \times 1.008 + 16$ or 18.016. To avoid confusion the formula agreed upon must always be specified, such as C for solid carbon, S for solid sulphur, and H_2O for liquid water. Consequently, the *formula weights* of the solids C and S are their respective atomic weights, regardless of what may eventually be found true regarding their molecular weights. Similarly, by using the formula weight of H_2O, the question of what the molecular weight of liquid water or ice may be does not enter the problem. Some scientists believe that liquid water is probably made up of a mixture of the molecules H_2O, $(H_2O)_2$, and $(H_2O)_3$, but in any case the formula weight of H_2O is 18.016.

The *units* in which the formula weights are expressed depend upon the system of units used. Thus, one *gram formula weight* (g.f.w.) of CO would be 28 *grams* of this substance; and similarly one *pound formula weight* (lb.f.w.) of CO would be 28 *pounds* of it.

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[5] In certain gases such as mercury vapor and the rare gases of the atmosphere, the molecules are monatomic.

[6] For further information on this subject see "X-Rays and Crystal Structure," by Sir William H. Bragg.

(d) **Chemical Equations** express a number of important facts regarding combustion. In such equations, (1) the *number of atoms* of the chemical elements must be the same before and after the reaction; (2) for each substance involved, the *relative weight*, in pounds or grams, entering into combination is represented by the product of its formula weight and its formula coefficient; and (3) the *relative volumes of gases* entering into combination are shown by their respective formula coefficients in the equation. For example, the chemical equation $2CO + O_2 \rightarrow 2CO_2$ signifies (1) that 2CO containing 4 atoms combine with 2 atoms of oxygen and there results a quantity of CO_2 containing 6 atoms; (2) that 2×28 lb. (or grams) of CO combine with 1×32 lb. (or grams) of O_2 to yield 2×44 lb. (or grams) of CO_2; (3) and, assuming that each substance is a gas, that two volumes of CO combine with one volume of oxygen and produce two volumes of CO_2. All these facts are illustrated by the equations given in Table XXXVI for the principal reactions encountered in combustion problems.

If the pound (or gram) **formula weight** be chosen as the unit of weight of a substance (whether solid, liquid or gas), the coefficient of the formula of that substance in a chemical equation then shows the number of such units of weight that are involved. For this reason the formula weights are convenient units in which to express the weights of substances in chemical reactions.

When *gases alone* are involved in a chemical equation, the formula coefficient of each gas expresses its relative volume because the formula of a gas represents a molecule of the substance and the same number of molecules occupy equal volumes when at the same temperature and pressure (Avogadro's Law). It follows that, when dealing only with gases, the volume of one *mole* (or mol) is the same for all gases; for a mole is m pounds (or grams) of a gas, where m is the molecular weight of that substance.[7] The number of molecules of a gas formed by a chemical reaction may or may not be the same as the number of molecules of the reacting gases; consequently, the gaseous products of combustion may or may not have the same volume as the mixture of gases before combustion, all measured at the same pressure and temperature. When solid and liquid fuels are involved, their volumes are very small relative to the volume of air, or oxygen (at atmospheric pressure and temperature) necessary for combustion, and therefore the volume of such fuels generally may be considered as negligible.

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[7] See p. 76, Part I.

TABLE XXXVI

SIGNIFICANCE OF CHEMICAL EQUATIONS FOR COMBUSTION

	Burning Solid Carbon to Carbon Dioxide	Burning Solid Carbon to Carbon Monoxide	Burning Carbon Monoxide to Carbon Dioxide	Burning Hydrogen to Water Vapor	Burning Solid Sulphur to Sulphur Dioxide
Equation.	$C+O_2 \rightarrow CO_2$	$2C+O_2 \rightarrow 2CO$	$2CO+O_2 \rightarrow 2CO_2$	$2H_2+O_2 \rightarrow 2H_2O$	$S+O_2 \rightarrow SO_2$
Relative number of atoms.	$1+2=3$	$2+2=4$	$4+2=6$	$4+2=6$	$1+2=3$
Relative weights.	$12+32=44$	$2\times12+32=2\times28$	$2\times28+32=2\times44$	$2\times2+32=2\times18$	$32+32=64$
Weights per pound of fuel.	$1+2\frac{2}{3}=3\frac{2}{3}$	$1+1\frac{1}{3}=2\frac{1}{3}$	$1+\frac{4}{7}=1\frac{4}{7}$	$1+8=9$	$1+1=2$
Relative volumes*.	$0+1 \rightarrow 1$	$0+1 \rightarrow 2$	$2+1 \rightarrow 2$	$2+1 \rightarrow 2$	$0+1 \rightarrow 1$

* For gases Avogadro's Law is applied, while for solids the volumes are taken as zero.

When the oxygen is supplied from the air, a large amount of nitrogen accompanies it; but the nitrogen generally enters into the combustion but very little, and it is therefore commonly spoken of as inert. In ordinary engineering applications the nitrogen term is simply added to both sides of the equation. However, the nitrogen increases the weight of the products of combustion very materially, and thus the maximum temperature attained is much less than when the same fuel is burned with oxygen only.

372. The Heating Value of Fuels.—(a) It is a simple matter to define the heating value of a fuel as the amount of heat that may be absorbed by cooling the products of combustion back to the original temperature of the fuel mixture; but when this definition is applied to different cases one soon discovers that a fuel may have various heating values, the value of each depending upon the composition of the products of combustion, the original temperature of the mixture, and whether the heat has been abstracted at constant pressure or at constant volume. All of these factors will not be fully discussed here,[8] but the heating values most commonly used will be considered.

(b) The **high or gross heating value** (H.H.V.) of a fuel is defined as the amount of heat absorbed from the products of complete combustion of a unit weight of the fuel when they are cooled to the initial temperature of the fuel mixture under such conditions that all of the water vapor formed from the combustion of hydrogen in the fuel is condensed. If the pressure is kept constant, the amount of heat abstracted is called the high heating value at *constant pressure*; whereas if the volume is kept constant, the result is called the high heating value at *constant volume.*

(c) The **low or net heating value** (L.H.V.) of a fuel is obtained when the products of complete combustion are cooled to the initial temperature of the fuel mixture *without* condensing any of the vapor formed by the burning of the hydrogen in the fuel. Just as in the previous case, the low value may be obtained at constant pressure or at constant volume.

The difference between the heating values obtained at constant volume and constant pressure is so small as to be negligible in most engineering applications.

(d) The *units* used to express the heating values of fuels are quite varied, and depend upon the unit of energy and the unit-amount of fuel chosen. The engineer in this country commonly prefers the B.t.u. per pound. However, for liquid fuels he may use the B.t.u. per gallon, and for gases the B.t.u. per cubic foot at some definite pressure and temperature. The standard pressure chosen is one atmosphere (760 mm. of Hg) and the standard temperature is often taken as 32, 60, or

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[8] For additional information see Bulletin 139, Univ. of Illinois.

68 deg. fahr., the latter value being specified in the A.S.M.E. Test Code on Definitions and Values.

(e) The **experimental determination** of the *high* and *low* heating values of a fuel may be obtained by means of suitable calorimeters. Thus, the *high heating value* at *constant volume* may be found with a bomb calorimeter using oxygen, by putting sufficient water in the bottom of the bomb to insure saturation at the beginning and end of the run. This procedure is commonly followed with solid fuels.

The *lower* heating value at *constant pressure* of a dry liquid or gaseous fuel may be obtained with a Junker's calorimeter using air, by obtaining, in addition to the other determinations, the actual weight of water condensed in the calorimeter and making correction therefor by subtracting the latent heat of that weight of steam for a temperature the same as the original one of the air and fuel. This procedure is readily followed with gaseous and liquid fuels, but this form of calorimeter is not well adapted for use with solid fuels.

The heating values at *constant pressure* are related as follows:

$$\text{H.H.V.} = \text{L.H.V} + 9\, w_H r \quad . \quad . \quad . \quad . \quad . \quad (516)$$

$$\text{L.H.V.} = \text{Cal. H.V.} - w_c r \quad . \quad . \quad . \quad . \quad . \quad (517)$$

in which r = latent heat of steam at the temperature to which the products of combustion are cooled in the calorimeter;

w_H = the fractional weight of hydrogen in the fuel;

w_c = weight of water condensed per pound of fuel;

Cal. H.V. = the calorimeter heating value for the case in which w_c lb. of water is condensed per pound of fuel.

(f) The **actual heating values,** gross and net, of various fuels are given in Table XXXVII. The values per gram-formula weight and per mole were taken from the International Critical Tables. All of the other heating values were calculated by the authors by using the conversion factors as given in the footnotes.

(g) For **gaseous fuels,** the table gives not only the heating values per pound, but also the values per *cubic foot* at an absolute pressure of 14.7 lb.per square inch and at a temperature of 64 deg. fahr.; and in addition it gives the value of the constant $R/144$. With such data available, the heating value per cubic foot of gas at any pressure, p, and absolute temperature, T, is found by the following relation:

$$\text{H.V. per cu. ft.} = \frac{\text{H.V. per lb.}}{(R/144)(T/p)} \quad . \quad . \quad . \quad . \quad . \quad (518)$$

TABLE XXXVII

COMBUSTION DATA

	Substance	Symbol	Atomic Weight	Formula	Formula Weight, (f)	$R \div 144$, in $\overline{V} = \left(\frac{R}{144}\right)\frac{T}{p}$ *	Reaction	High or Gross Heating Value at Constant Pressure of 1 Atmosphere and Initial Temperature of 18° C. or 64° F.					Low or Net Heating Value at 64° F., B.t.u. per Lb. ¶	Weight of Oxygen Required per Lb. of Fuel, Lb.	Air Required for Combustion		Specific Heats
								Abs. kj. per gram formula weight †	Kg. cal. per gram mole ‡	B.t.u. per lb. formula weight	B.t.u. per lb.	B.t.u. per cu. ft.			Lb. per lb. of fuel	Cu. ft. per cu. ft. of fuel	
Common Combustibles	Carbon (solid)	C	12.000	C	12.000	....	$C+O_2=CO_2$	395.0	....	169,800	14,150	......	14,150	2.67	11.5	......	c_p and c_v are very dependent on temp.; see p. 799, Appendix
							$2C+O_2=2CO$	110.6	....	47,520	3,960	......	3,960	1.33	5.75	......	
	Carbon Monoxide	...	....	CO	28.000	0.3840	$2CO+O_2=2CO_2$	....	....	122,280	4,367	319	4,367	0.57	2.46	2.38	
	Hydrogen	H	1.008	H_2	2.016	5.292	$2H_2+O_2=2H_2O$	286.2	....	122,980	61,000	323	51,500	8.00	34.5	2.38	
	Sulphur (solid)	S	32.065	S	32.065	....	$S+O_2=SO_2$	290	....	124,730	3,890	......	3,890	1.00	4.32	......	
	Methane (gas)	...	....	CH_4	16.032	0.67	$CH_4+2O_2=CO_2+2H_2O$	....	210.8	379,480	23,670	991	21,290	4.00	17.25	9.56	
	Acetylene (gas)	...	....	C_2H_2	26.016	0.411	$2C_2H_2+5O_2=4CO_2+2H_2O$	....	312.0	561,680	21,590	1,473	20,860	3.08	13.25	12.0	
	Ethylene (gas)	...	....	C_2H_4	28.032	0.384	$C_2H_4+3O_2=2CO_2+2H_2O$	....	332	597,640	21,320	1,558	19,960	3.43	14.8	14.3	
	Ethane (gas)	...	....	C_2H_6	30.048	0.356	$2C_2H_6+7O_2=4CO_2+6H_2O$	....	368.4	663,160	22,070	1,739	20,170	3.73	16.1	16.8	
	Benzene (gas)	...	....	C_6H_6	78.048	0.137	$2C_6H_6+15O_2=12CO_2+6H_2O$	....	778.2	1,400,180	17,940	3,673	17,210	3.08	13.26	36.0	
	Benzene (liq.)	...	....	C_6H_6	78.048	....	$2C_6H_6+15O_2=12CO_2+6H_2O$	....	783.4	1,410,330	18,070	......	17,340	3.08	13.26	......	
	Hexane (gas)	...	....	C_6H_{14}	86.112	0.124	$2C_6H_{14}+19O_2=12CO_2+14H_2O$	....	993.9	1,788,550	20,770	4,700	19,220	3.53	15.21	45.6	
	Hexane (liq.)	...	....	C_6H_{14}	86.112	....	$2C_6H_{14}+19O_2=12CO_2+14H_2O$	....	990.6	1,782,520	20,700	......	19,150	3.53	15.21	......	
	Octane (liq.)	...	....	C_8H_{18}	114.144	....	$2C_8H_{18}+25O_2=16CO_2+18H_2O$	....	1305	2,349,080	20,580	......	19,080	3.5	15.1	......	
	Methyl Alcohol (liq.)	...	....	CH_4O	32.032	....	$2CH_4O+3O_2=2CO_2+4H_2O$	....	170.9	307,510	9,600	......	8,410	1.5	6.46	......	
	Ethyl Alcohol (liq.)	...	....	C_2H_6O	46.048	....	$C_2H_6O+3O_2=2CO_2+3H_2O$	....	328	590,330	12,820	......	11,580	2.09	9.00	......	
Non-Combustible gases	Oxygen	O	16.000	O_2	32.000	0.3347											
	Nitrogen	N	14.008	N_2	28.016	0.3840											
	Air ‖	...	....	....	....	0.3701											
	Carbon Dioxide	...	....	CO_2	44.000	0.2450											
	Sulphur Dioxide	...	....	SO_2	64.065	0.167											
	Steam	...	....	H_2O	18.016	0.595§											

* For a gaseous fuel this equation gives the specific volume ($\overline{V}$) in cu. ft. per lb., for any pressure (p) in lb. per sq. in. and absolute temperature (T) in °F.

† From the International Critical Tables, 1929, Vol. V, p. 176. (1 abs. k. joule per gram formula weight $= 429.7/f$ B.t.u. per lb.)

‡ From the International Critical Tables, 1929, Vol. V, p. 163. (1 kg. cal. per gram mole $= 1800/f$ B.t.u. per lb.)

¶ These net values are also at constant pressure.

‖ Air is a mixture, and may be considered as 20.9% O_2 and 79.1% N_2 by volume, or 23.2% O_2 and 76.8% N_2 by weight.

§ This value is accurate only for saturated and superheated steam with $p < 2$ lb. per sq. in. abs.

(**h**) Special attention is called to the fact that the table shows the **high heating value of carbon** to be 14,150 B.t.u. per pound, which is nearly 3 per cent less than the values that have been commonly used heretofore. However, the old value of about 14,500 is based chiefly on the experiments of Favre and Silbermann [9] made in 1852 and other early investigators, whereas the new value is the result of many very careful experiments made by Roth and Naeser in 1925–26. Similarly, the new gross value as given in this table for **hydrogen** is 61,000 B.t.u. per pound as contrasted with the old value of 61,550, which is about 0.9 per cent higher than the new value. By using these new values in calculating the heating values of hydrocarbons one will obtain a very satisfactory check on the experimentally determined heating value of such fuels as gasoline (octane) and acetylene.

(i) **The relation between the heating values** of carbon and carbon monoxide, and also the heat developed when carbon is burned to carbon monoxide only, are important in combustion engineering. From the information contained in Tables XXXVI and XXXVII, it is found that,

$$1 \text{ lb. C} + 2\tfrac{2}{3} \text{ lb. } O_2 \rightarrow 3\tfrac{2}{3} \text{ lb. } CO_2 + 14{,}150 \text{ B.t.u.} \quad . \quad . \quad (519)$$

$$1 \text{ lb. C} + 1\tfrac{1}{3} \text{ lb. } O_2 \rightarrow 2\tfrac{1}{3} \text{ lb. CO} + 3{,}960 \text{ B.t.u.} \quad . \quad . \quad (520)$$

$$1 \text{ lb. CO} + \tfrac{4}{7} \text{ lb. } O_2 \rightarrow 1\tfrac{4}{7} \text{ lb. } CO_2 + 4{,}367 \text{ B.t.u.} \quad . \quad . \quad (521)$$

Then it follows that if 1 *pound of carbon* is burned to CO_2 according to Eq. (519), or according to Eqs. (520) and (521) combined, the heat given off will be the same. Thus, from (520) for 1 lb. of C and from (521) for $2\frac{1}{3}$ lb. of CO,

$$3960 + (2\tfrac{1}{3})\,(4367) = 3960 + 10{,}190 = 14{,}150 \text{ B.t.u. per lb. C.}$$

These equations also show that if a pound of carbon is burned to CO and the heat so developed (3960 B.t.u.) *is allowed to waste,* as is often done with producer gas, the maximum amount of heat that can be obtained from burning the resulting *cold* CO to CO_2 is 10,190 B.t.u., which is only 72 per cent of that obtainable by not allowing the CO to cool off before burning.

If, for any reason, only the fractional part, x, of carbon is burned to CO_2 and the rest, $1 - x$, is burned to CO, the heat developed, in B.t.u. per pound of carbon, is

$$Q_x = 14{,}150x + 3960\,(1 - x) \quad . \quad . \quad . \quad . \quad . \quad . \quad (522)$$

(j) When there is *less than enough oxygen* to burn carbon to carbon dioxide, both CO_2 and CO will be formed and the relative amounts of each can be determined in the following manner: Assume first that there is 1 per cent deficiency in the oxygen supply needed to form CO_2 and that in consequence 99 per cent of the carbon is burned to CO_2 and 1 per cent remains C; then assume that this 1 per cent of C combines with some of the CO_2 according to the equation $C + CO_2 = 2CO$; thus it is seen that there is finally 2 per cent of the carbon present in CO and 98 per cent in CO_2. In general, then, if there is *y per cent deficiency of oxygen* there will be $2y$ per cent of the carbon burned to carbon monoxide instead of to CO_2. Hence, it follows that with y per cent deficiency of oxygen *there is* $(2 \times 0.72)y$ *per cent, or* $1.44y$ *per cent less heat developed* than if all the carbon were burned to CO_2. The great importance of having a sufficient supply of oxygen is thus apparent.

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[9] See p. 118, Bulletin 139, Univ. of Illinois Eng. Exp. Station, by Goodenough and Felbeck.

The discussion in the preceding paragraph presupposes that the oxygen supply is at least sufficient to burn all the carbon to CO—that is, that the deficiency is not more than 50 per cent on a basis of combustion to CO_2. Should y be greater than 50 per cent, some of the carbon will not be burned at all. The percentage not affected will be $2(y - 50)$.

(**k**) It is sometimes possible to *reverse chemical reactions*—in the present case, for instance, to break up one of the oxides into the original elements. When this is done the same amount of heat will be absorbed during the decomposition as was originally liberated during combination.

Thus, if it is possible to break up the quantity of CO_2 containing a pound of carbon into the elements C and O, it will require an expenditure of 14,150 B.t.u. Similarly, it will require 10,190 B.t.u. to reduce to the monoxide CO an amount of CO_2 containing 1 pound of carbon, and 3960 B.t.u. per pound of carbon will be consumed in separating CO into its elements.

(**l**) The **heating values based on the formula weight** are often used by scientists, and engineers need to know how to convert from one unit to another. If each term in Eqs. (519), (520), and (521) be multiplied by the number expressing the formula weight of the fuel the resulting heating values, will be, respectively,

$$\text{C to } CO_2,\ 12 \times 14{,}150 = 169{,}800\ \textit{B.t.u. per lb. f.w.}\text{ of C} \quad (a)$$

$$\text{C to CO},\ 12 \times 3{,}960 = 47{,}520\ \textit{B.t.u. per lb. f.w.}\text{ of C} \quad (b)$$

$$\text{CO to } CO_2,\ 28 \times 4{,}367 = 122{,}280\ \textit{B.t.u. per lb. f.w.}\text{ of CO} \quad (c)$$

A similar relation holds for other fuels, as may be seen from Table XXXVII. Note should be made of the fact that the sum of the heating values of (*b*) and (*c*) equals that of (*a*).

When transforming heating values from one unit of weight to another involving the formula weight, attention needs to be paid to the value used for atomic weights. Thus, for example, consider the case of hydrogen (H_2) with a heating value of 61,000 B.t.u. per pound: The corresponding heating value in B.t.u. *per pound formula weight* becomes $61,000 \times 2.016 = 122,976$. If the formula weight of H_2 had been taken as 2 in determining the B.t.u. per pound formula weight, the value resulting for the latter quantity would have been 122,000, which is 976 B.t.u. less than that obtained by using 2.016. While this difference by itself may appear large, it is but little more than three-quarters of 1 per cent of the total, and may or may not be considered of importance, depending upon the degree of accuracy required for the particular calculation involved. Generally, the experimental data used in engineering combustion calculations are not accurate to 1 per cent.

The atomic and formula weights of various substances needed in the usual engineering applications of combustion are given in Table XXXVII.

373. Oxygen and Air Required for Combustion.—(a) The **weight of oxygen** required to burn a unit weight of fuel can be calculated when one knows the composition of the fuel and its chemical reaction with oxygen. From the chemical equations given in Table XXXVI it may be seen that each pound of carbon burned to CO_2 requires $2\frac{2}{3}$ pounds of oxygen, and only $1\frac{1}{3}$ pounds if burned to CO; also 8 pounds of oxygen are required to burn each pound of hydrogen; and 1 pound of oxygen is needed for each pound of sulphur. Since these elements constitute our chief fuels, the general expression for the *weight of oxygen* required to burn a pound of fuel to CO_2, H_2O, and SO_2, is

$$w_O = 2\tfrac{2}{3}w_C + 8(w_H - w_O/8) + w_S \text{ lb.}, \quad . \quad . \quad . \quad . \quad (523)$$

in which w_C, w_H, w_O, and w_S represent the *fractional parts by weight* of carbon, hydrogen, oxygen, and sulphur in the fuel. In this and the following equations the term $(w_H - w_O/8)$ assumes that all of the oxygen represented by w_O is free to unite with the hydrogen.

(b) Since air contains only 23.2 per cent oxygen by weight, the **weight of air** needed to supply the oxygen of Eq. (523) becomes

$$w_a = 11.5w_C + 34.5(w_H - w_O/8) + 4.32w_S \text{ lb.} \quad . \quad . (524)$$

(c) The **volume of air** required *per pound* of fuel is

$$V_a = w_a\left(\frac{53.3}{144}\right)\left(\frac{T_a}{p_a}\right) = 0.37w_aT_a/p_a \text{ cu. ft.}, \quad . \quad . \quad (525)$$

in which T_a and p_a represent the absolute temperature and pressure of the air for which the volume is desired.

(d) For a *gaseous fuel* the volume of the air required per cubic foot of gas, measured at the same temperature and pressure as that of the air, would be

$$V'_a = w_a R_a / R_g \text{ cu. ft.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (526)$$

where R_a and R_g represent the constants of air and the gaseous fuels, respectively.

In all of these equations, and also in Table XXXVII, the amount of air is that which is just sufficient to complete the combustion; or, in other words, *no excess* or *no deficiency* of air has been considered.

(e) A **deficiency of air** supplied to a fuel made up of the usual elements is considered to affect only the combustion of the carbon because the hydrogen and sulphur both combine with oxygen so much more readily than does carbon. For each pound of carbon, of which only the fraction x is burned to CO_2 because of lack of oxygen and the remainder is burned to CO, the weight of air used by the carbon is

$$w_{a_x} = 11.5x + 5.75(1 - x) \text{ lb.} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (527)$$

374. Some Actual Reactions.—(a) In the actual combustion processes the reactions may not be carried out exactly in accordance with the simple equations given. Often, before the final product is obtained, a series of reactions may occur, and in burning carbon, for example, primary transitory complex compounds may be formed which decompose later into CO and CO_2 in amounts which depend on the conditions under which the reaction takes place.

(b) If **coal is burned in a furnace,** the combustion takes place in several steps and in several zones. When fresh coal is spread on the top of the fuel bed the volatile matter will be driven off in what constitutes the *distillation zone.* To burn the volatile fuel, *secondary* air must be mixed with it, at the proper temperature, in the *combustion zone* above. The coke that remains is added to the active fuel bed in which the carbon is burned in the following manner: In the lower three to four and one-half inches of the active bed, which portion constitutes the *oxidizing zone,* the carbon combines exothermically with the oxygen of the ascending "*primary*" air to form CO_2, little or no CO being produced in this region. Then, as this CO_2 passes up through the hot fuel, it enters the *reducing zone* in the fuel immediately above, and some of it reacts endothermically with the incandescent carbon there to form CO, but never in an amount exceeding that corresponding to the "equilibrium ratio"[10] of CO to CO_2 for the existing temperature and pressure. If the temperature is high in this zone, and if the time of contact and thickness of the incandescent bed of carbon are sufficient, most of the CO_2 will be reduced to CO. Finally, the resulting gas mixture passes through the distillation zone into the *combustion space*, or zone, above the fuel bed where the CO is burned to CO_2 by means of secondary air supplied at this point. The net amount of heat evolved from this series of carbon reactions is the same as that which results from the simple reaction $C + O_2 = CO_2$.

(c) The **process of manufacturing producer gas** is another example of complicated combustion. In this process, air is passed up through a very deep column of fuel in such quantity as to react with the carbon to form carbon monoxide (CO), which is the principal combustible in this gas. The reaction, however, is not all carried out according to the simple equation $2C + O_2 = 2CO$. In the actual process CO_2 is first formed (with the evolution of heat) in the lower part of the producer, the simple elements of which are shown in Fig. 521; and later this gas is converted, endothermically, in the upper part of the fuel bed to the desired CO; but this reaction is never quite complete. Experiments show that when a stream of carbon dioxide passes through a tube containing hot carbon there will always be a certain amount of carbon dioxide mixed with the issuing carbon monoxide, and this also occurs in the producer. The relative percentages of these two constituents depend on such factors as (*a*) the existing temperature, (*b*) the pressure, (*c*) the time of contact, and (*d*) the surface condition of the carbon. Other things being equal, and with adequate time allowance, there are definite proportions of carbon monoxide and dioxide which will be in equilibrium

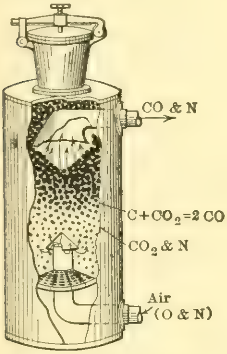

FIG. 521.—Gas Producer.

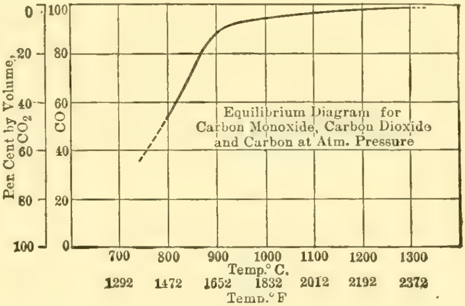

FIG. 522.

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[10] This term will be defined in the next paragraph.

with carbon at each given temperature and pressure, as shown in Fig. 522 for one pressure. The corresponding ratio between these constituents is termed the **equilibrium ratio.** How the proportions vary with time when the carbon is that in charcoal, coke, and anthracite coal, are illustrated, for different temperatures, by the respective curves in Figs. 523 and 524. To obtain the maximum amount of CO in the gas, there should be within the active portion of the producer (1) a high temperature, (2) long contact between the gases and the hot carbon, (3) maximum porosity of fuel, (4) minimum size of particles, and (5) the correct amount and distribution of air.[11] In order to avoid trouble from fusion of the ash in the base of the producer, steam (or occasionally CO_2 from flue gas) is introduced with the air, thus reducing the temperature somewhat. The gas thus produced contains CO, CO_2, water-gas, the volatile matter that is distilled from the coal, and nitrogen from the air used.

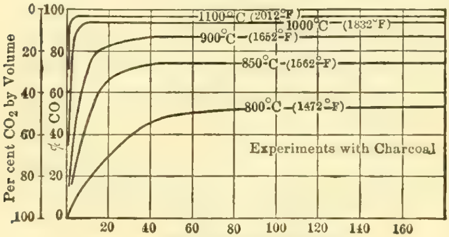

FIG. 523.

(**d**) The formation of **water gas** is still another example. In this process steam is blown through a bed of incandescent carbon and thus the principal constituents of this gas, carbon monoxide and hydrogen, are produced mainly as given by Eq. (*a*) following:

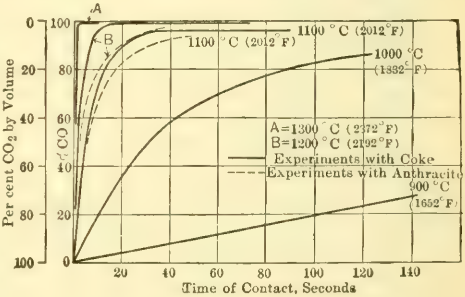

FIG. 524.

$$C + H_2O = CO + H_2 - 75{,}460 \text{ B.t.u.} \quad . \quad . \quad . \quad . \quad (a)$$

$$C + 2H_2O = CO_2 + 2H_2 - 76{,}160 \text{ B.t.u.} \quad . \quad . \quad . \quad (b)$$

$$CO + H_2O = CO_2 + H_2 - 700 \text{ B.t.u.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (c)$$

$$C + CO_2 = 2CO - 74{,}760 \text{ B.t.u.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (d)$$

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[11] Bu. of Mines Bul. 7, "Essential Factors in the Formation of Producer Gas" (1911); "Fuel Saving in Modern Gas Producers and Industrial Furnaces," by W. B. Chapman, Mechanical Engineering, Nov., 1921, p. 717; Bul. 30, U. of Ills. Engineering Exp. Station. Also see Bu. of Mines Bul. 13, and Tech. Paper 63.

The reactions (*b*), (*c*), and (*d*), however, are also taking place simultaneously and all four reactions must be considered together. The equilibrium ratio between the substances on the two sides of each of these equations depends on the temperature existing and the amount of excess steam used, and these factors fix the composition of the gas formed. In these equations the heat developed (or absorbed) is given in B.t.u. per pound formula weight.

The amount of heat absorbed by the reaction (*a*) is found by the aid of Table XXXVII, from which it may be seen that 122,980 B.t.u. per pound formula weight of hydrogen are absorbed to form the H_2 of this equation. This table also shows that the formation of the CO *develops* 47,520 by combining with the same amount of oxygen that was given up by the dissociation of the H_2O; hence the net heat *developed* by the reaction (*a*) is $47{,}520 - 122{,}980 = -75{,}460$ B.t.u. per lb. f. w. (or 12 lb.) of carbon.

For the reaction (*b*), the dissociation of $2H_2O$ *absorbs* 245,960 B.t.u. and combining the resulting oxygen with the carbon to form CO_2 yields 169,800; therefore, the net heat developed is $169{,}800 - 245{,}960 = -76{,}160$ B.t.u. per lb. f.w. of carbon.

Similarly for reaction (*c*), the heat *developed* by the burning of 1 mole of CO is 122,280 B.t.u. and that *absorbed* in liberating the necessary oxygen from H_2O is 122,980, thus giving a net amount of -700 B.t.u. developed.

Also, for reaction (*d*), the oxygen to burn the carbon comes from dissociating the CO_2, thus yielding $47{,}520 - 122{,}280$, or $-74{,}760$ B.t.u.

375. Combustion of Hydrocarbons.—(**a**) As already noted, combustibles composed of hydrogen and carbon in chemical combination are known as "hydrocarbons" (C_nH_m), of which there are many kinds which differ from each other in the relative proportions of C and H contained. At ordinary temperatures some hydrocarbons are gaseous, some are liquids, and others are solids. With complete combustion, all ultimately burn to form H_2O and CO_2, but the process is often very complicated, for a series of reactions may go on before it is completed. The final amounts of H_2O and CO_2 produced, however, are the same as those shown by the simple reaction formulas.

(**b**) If the hydrocarbon contained w_C pounds of carbon and w_H pounds of hydrogen per pound of fuel, and if these elements were burned separately and all of the H_2O were condensed, then the combustion would liberate an amount of heat equal to

$$\text{H.H.V.}' = 14{,}150w_C + 61{,}000w_H \text{ B.t.u. per lb. of fuel.} \quad . \quad (528)$$

on the basis of higher heat value of the hydrogen; but the results of such calculations seldom check with experimentally determined values. This is because the hydrocarbon is already a chemical compound which first must be broken up to enable the individual elements to combine with the oxygen, this separation being accompanied by the absorption or liberation of heat, which would diminish or increase the net amount evolved. Many empirical formulas have been developed to take account of such effects, but none is entirely satisfactory for all cases. In many instances the approximate calculations given by Eq. (528) are sufficiently exact, but when greater accuracy is desired the experimentally determined values should be used. These values are given in the tables in reference books. The most authentic values are contained in the International Critical Tables.

(**c**) If the hydrocarbon is of the composition C_nH_m, the *weights of oxygen and air* required per pound for complete combustion are very closely,

$$w_O = (32n + 8m) \div (12n + m) \text{ lb.} \qquad (529)$$

$$w_a = 4.32 \times w_O \qquad (530)$$

The only inaccuracy involved in these equations is the slight one due to using the approximate value of the atomic weight of hydrogen as 1.

(**d**) Since the *heating value per cubic foot* of gas can be obtained from the heating value per pound by dividing the latter by the specific volume, it follows that for each gas there will be as many different heating values as there are temperature and pressure combinations; hence the pressure and temperature under which the volumetric measurement of the gas is made should always be stated. When the fuel contains hydrogen or hydrocarbons, discretion must be exercised in each case in deciding whether the higher or lower heating value should be used.

(**e**) The heating values of some of the gases do not differ so widely on the volume basis as they do when expressed in terms of weight. When mixed with just the correct amount of air for complete combustion, the heating values of the mixtures of all the gases tabulated differ but little.

This may be illustrated by making a *comparison of carbon monoxide and hydrogen* as fuels. From Table XXXVII it may be seen that the heating value of a *pound* of hydrogen is about 14 times as great as that of an equal weight of carbon monoxide, but for the same *volume* the difference is only a little over *one* per cent.

Both H_2 and CO require theoretically the same volume of air per cubic foot for combustion, but in the actual process it may be necessary to have more excess air with hydrogen than with carbon monoxide. These relations are of particular importance in engineering, because:

(1) There are a large number of commercial gases containing both hydrogen and carbon monoxide, and it is possible to regulate the relative proportions of the two to a certain extent.

(2) It is generally the volume of the gas which is to be handled, and not its weight, which determines the dimensions and cost of apparatus and cost of operation.

These factors must be considered when comparing the sizes of gas engines necessary to produce a given power with these respective gases or their mixtures.

376. Combustion of Sulphur.—Sulphur burned in oxygen forms either SO_2 or SO_3 depending on the amount of oxygen. For each pound of S one pound of O is needed to form two pounds of SO_2. To supply this oxygen approximately 4.32 pounds of air are required. Although this reaction is accompanied by the liberation of about 3900 B.t.u. per pound of S, this element is not a desirable combustible because SO_2 and also SO_3, when combined with water, form acids that actively attack iron, copper, and many other engineering materials. Lead and bitumin, however, are immune to these acids; hence passages for flue gases which come from fuels containing sulphur are often lined or coated with such substances if there is a possibility of the presence of moisture.

377. Chemical Equilibrium and Dissociation.—(**a**) When, in Sect. **374,** the actual combustion processes were considered, it was observed that some of the reactions are never completed in one direction only. This matter will be given further consideration in this section.

(**b**) Many careful experiments have shown that in a mixture of gases, in which chemical action is possible, the tendency of this action to proceed in a certain direction is dependent upon the temperature and partial pressures of the constituents of this mixture. **Chemical equilibrium** is the state reached by the reacting substances when the rate of reaction in one direction is just balanced by the rate of reaction in the opposite direction. The term "*dissociation*," as ordinarily applied to combustion, simply means that the products of combustion are dissociating into the former constituents just as fast as they are formed. In other words, the effect of dissociation is the same as though the combustion had *halted* entirely, but the word "dissociation" does not fully express the facts because both reactions are taking place instead of a static condition existing. Consequently, the use of the term *equilibrium* in such cases is far better than "dissociation." Since chemical equilibrium during combustion depends so largely upon temperature, and somewhat on pressure, it may happen that the products of combustion in an engine (or even in a boiler furnace) may reach such a high temperature that *equilibrium* is attained before all of the fuel is consumed, and "dissociation" is then commonly said to occur.

The *method of calculating* the equilibrium curves for various reactions cannot be given here, but may be found in textbooks on Physical Chemistry and it is also given by Goodenough and Felbeck.[12]

(c) The **significance of chemical equilibrium,** as involved in combustion, can perhaps be best understood by means of an illustration. For example, consider the burning of carbon monoxide with just sufficient oxygen, or air, to form carbon dioxide if equilibrium is not reached before all of the CO is burned. Let the combustion take place at constant volume without any transfer of heat to or from the substances, and assume the initial absolute temperature of the mixture to be 520 deg. fahr. and the initial pressure one atmosphere. The equilibrium curves, using oxygen and air, are clearly shown in Fig. 525, as obtained from the data of Goodenough and Felbeck. In this figure there are also drawn the two temperature curves (marked T), one for the fuel and oxygen, and the other for fuel and air. Both of these temperature curves are, by hypothesis, assumed to start with $T = 520$ when none of the CO has been burned. The temperature naturally rises more slowly if air is used instead of pure oxygen, because of the large quantity of nitrogen that must be heated up. When 40 per cent of the CO has burned to CO_2, the two temperature curves show that the respective temperatures would be 5400 and 3100 deg. fahr. absolute. By following these two tem-

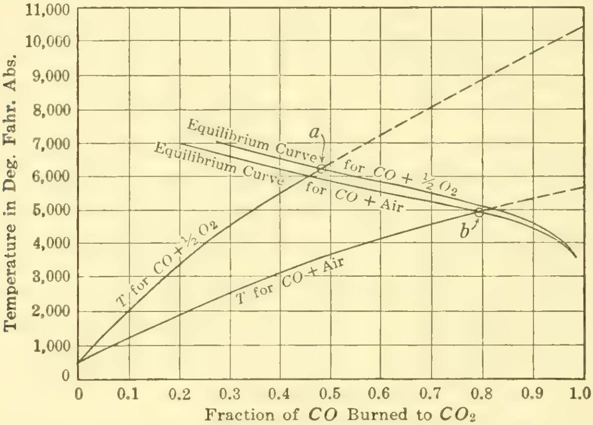

FIG. 525.—Combustion of Carbon Monoxide at Constant Volume.

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[12] See "An Investigation of the Maximum Temperatures and Pressures Attainable in the Combustion of Gaseous and Liquid Fuels," Bulletin No. 139, Univ. of Illinois.

perature curves until each intersects its own equilibrium curve, **the maximum temperature attainable** by the combustion of each mixture under the specified conditions is determined. Thus for the carbon monoxide and oxygen mixture this intersection is seen to be at *a*, which means that the maximum temperature attainable is about 6100 F. abs. and that only about 47 per cent of the CO has been burned. On the other hand, the lower temperature of the mixture with air permits about 80 per cent of the CO to be burned before equilibrium is reached, but the corresponding temperature is only about 4900 F. abs., as shown by the intersection of the two curves at *b*.

The temperature curves in Fig. 525 are continued as broken lines beyond the equilibrium curves to show what the temperatures would become in each case if all of the fuel could be burned to CO_2 regardless of how high the temperature might become. The temperature curves are far from straight lines because the specific heats increase so rapidly at high temperatures, as shown by Table E and Plate VIII in the Appendix.

(**d**) As *another illustration* to show the significance of the equilibrium curves of Fig. 525, suppose the combustion of CO with air has proceeded adiabatically until 80 per cent of the fuel has been burned and therefore equilibrium reached with a temperature of about 4900 F. abs. If this mixture were now permitted to transfer some heat to another body, or transform some heat into mechanical energy, as in the cylinder of a gas engine, the temperature would drop and, consequently, the combustion would proceed until at a temperature of 3500 F. abs. about 99 per cent of the CO would be burned to CO_2.

If the temperature were reduced still more, the percentage of CO burned to CO_2 would increase slowly until at about 2500 deg. fahr. abs. there would be only about 0.01 per cent of CO remaining. Adiabatic combustion of the same mixture at *constant pressure* would give equilibrium and temperature curves similar to, but somewhat lower than, those for constant volume.

(**e**) The *experimental determinations* of the amount of dissociation of CO_2 and H_2O that take place when these substances are heated to various temperatures under a pressure of one atmosphere are shown in Table XXXVIII.

(**f**) The *effect of pressure and temperature* on the dissociation of H_2O and CO_2 is shown by the curves [13] in Fig. 526. From these curves and also from Table XXXVIII it may be seen that at temperatures less than

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[13] Calculated for the authors by Mr. J. M. Clark, a graduate student in Chemistry at Cornell University.

TABLE XXXVIII *

EXPERIMENTAL DETERMINATIONS OF THE DISSOCIATION OF CO_2 AND H_2O AT ONE ATMOSPHERE PRESSURE

CO_2			H_2O		
Absolute Temperature, Degrees Fahr.	Per Cent Dissociated	Investigator	Absolute Temperature, Degrees Fahr.	Per Cent Dissociated	Investigator
2340	0.0041	Nernst and von Wartenburg	2385	0.0033	Langmuir
			2439	0.0049	
			2507	0.0068	
2511	0.0142	Langmuir	2515	0.0073	Nernst and von Wartenburg
2520	0.01	Nernst and von Wartenburg	2581	0.0103	Langmuir
	0.02		2614	0.0137	
			2624	0.0147	
			2653	0.0140	
2597	0.025	Langmuir			
			2664	0.0189	Nernst and von Wartenburg
2660	0.029	Nernst and von Wartenburg			
	0.035		2756	0.0255	Langmuir
			2790	0.0287	
2666	0.0281	Langmuir	2810	0.034	Nernst and von Wartenburg
2696	0.0471				
2817	0.064				
			3069	0.102	Lowenstein
			3209	0.182	
3272	0.45	Lowenstein	3353	0.354	
			3542	0.518	
4037	4.5	Emich	3879	1.180	von Wartenburg
4361	10.0–11.0		4063	1.770	
			4140	2.60	Nernst
4752	21.0	Bjerrum			
5182	51.7		4756	4.3	Bjerrum
5220	49.2		4856	7.5	
5301	64.7		4970	6.6	
5609	76.1		5101	9.8	
			5272	11.1	

* See pages 124 and 127 of Bulletin No. 139 of Engineering Experimental Station, University of Illinois, by Goodenough and Felbeck.

about 2500 deg. fahr. the dissociation of both of these substances is negligible even at low pressures; but higher temperatures and low pressures cause a rapid increase, especially so for carbon dioxide above 3500 deg. fahr.

(g) *Dissociation of* CO_2 *and* H_2O *in the internal combustion engine* causes a reduction in the maximum pressure and temperature, and this effect is generally more pronounced in the Otto than in the Diesel engine because the latter uses more air per unit weight of fuel, especially at light loads. The calculations of Goodenough and Baker[14] show that an *ideal Otto engine* with a compression ratio of 5 to 1, and using Octane (C_8H_{18}) as a fuel with just sufficient air to burn the fuel completely, would have about 81 per cent of the CO burned to CO_2 and about 97 per cent of the H_2 burned to H_2O at the beginning of expansion; and at the end of expansion these percentages would rise to 97.6 and 99.5, respectively. Higher compression ratios will increase these values somewhat; or, in other words, the dissociation will be slightly less with the increased compression ratio, provided the amount of air originally mixed with the gasoline is 100 per cent. This quantity of air means just enough to allow all CO to be burned to CO_2 and all hydrogen to be burned to H_2O, assuming the resulting temperature and pressure will not cause chemical equilibrium to be established before this has been accomplished. On the other hand, if the air supplied is only 75 per cent, the dissociation of CO_2 at the end of the constant volume process is slightly increased by the higher compression ratios.

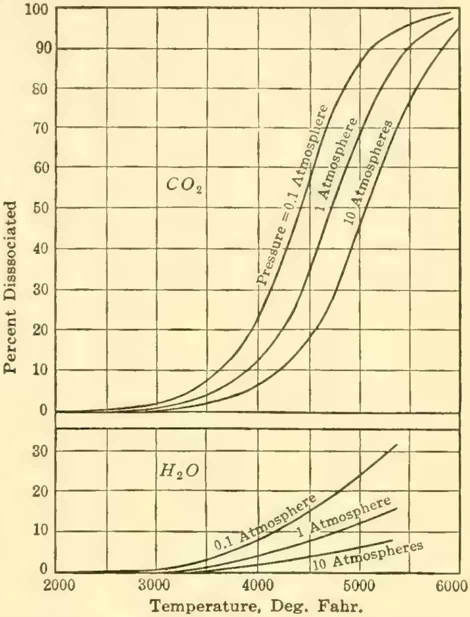

FIG. 526.—Effect of Temperature and Pressure on Dissociation of CO_2 and H_2O.

The various temperatures and the amounts of "dissociation" that would occur under adiabatic conditions of combustion, such as prevail in the ideal Otto and Diesel engines, may be calculated for different compression ratios, different fuel mixtures and different initial temperatures. Such calculations are very complex, but may be found in the excellent bulletin of Goodenough and Baker. Some of their results are given in Table XXXIX, which contains much interesting information.

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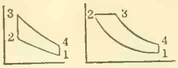
[14] Bulletin No. 160, Engineering Experiment Station, University of Illinois.

TABLE XXXIX

THE TEMPERATURES AND AMOUNTS OF DISSOCIATION IN IDEAL ENGINES

(As Calculated by Goodenough and Baker in Bulletin No. 160, University of Illinois)

Cycle	Compression Ratio	Fuel	Per Cent of Air*	p_1	 Absolute Temperature Deg. Fahr.				x = fraction of CO transformed into CO_2, or $1-x$ = fraction of CO_2 dissociated. y = fraction of hydrogen transformed into H_2O, or $1-y$ = fraction of H_2O dissociated.			
					T_1	T_2	T_3	T_4	x_3	x_4	y_3	y_4
Otto	5.0	Gasoline (Octane) (C_8H_{18})	75	14.7	622	1029	4753	3176	0.4070	0.4736	0.8316	0.7733
			90	....	632	1062	5067	3549	0.6911	0.7583	0.9462	0.9370
			100	....	638	1081	5053	3736	0.8113	0.9756	0.9712	0.9950
			110	....	635	1085	4940	3521	0.8838	0.9965	0.9830	0.9992
			125	....	630	1088	4721	3218	0.9440	0.9995	0.9919	0.9998
			150	....	630	1101	4351	2830	0.9835	1.0000	0.9975	1.0000
Otto	6.5	C_8H_{18}	75	14.7	594	1066	4810	2995	0.4058	0.4875	0.8333	0.7620
			90	....	601	1098	5121	3355	0.6925	0.7644	0.9474	0.9317
			100	....	605	1117	5108	3553	0.8150	0.9858	0.9724	0.9969
			110	....	600	1119	4995	3327	0.8868	0.9987	0.9838	0.9997
			125	....	595	1123	4774	3029	0.9453	1.0000	0.9927	1.0000
			150	....	595	1139	4406	2655	0.9835	1.0000	0.9988	1.0000
Otto	8.0	C_8H_{18}	75	14.7	577	1105	4829	2837	0.4050	0.5015	0.8338	0.7487
			90	....	582	1137	5165	3202	0.6943	0.7689	0.9480	0.9278
			100	....	586	1174	5150	3405	0.8178	0.9918	0.9733	0.9980
			110	....	585	1167	5040	3187	0.8870	0.9995	0.9840	0.9998
			125	....	580	1173	4821	2890	0.9454	1.0000	0.9924	1.0000
			150	....	580	1183	4451	2527	0.9825	1.0000	0.9971	1.0000
Otto	5.0	Benzene (C_6H_6)	75	14.7	636	1096	5138	3476	0.4378	0.4739	0.8622	0.8020
			100	....	655	1146	5237	3977	0.7669	0.9559	0.9649	0.9918
			110	....	648	1141	5139	3768	0.8385	0.9909	0.9766	0.9981
			125	....	642	1139	4931	3438	0.9108	0.9985	0.9873	0.9996
			150	....	639	1142	4560	3013	0.9700	1.0000	0.9943	1.0000
	6.5	....	100	....	615	1180	5313	3801	0.7704	0.9724	0.9660	0.9945
	8.0	....	100	....	595	1226	5354	3660	0.7681	0.9805	0.9659	0.9959
Diesel	8.0	Kerosene ($C_{12}H_{26}$)	100	14.7	602	1360	4854	4286	0.8183	0.9162	0.9710	0.9848
			200	....	585	1323	3556	2595	1.0000	1.0000	1.0000	1.0000
			300	....	590	1333	2888	1885	1.0000	1.0000	1.0000	1.0000
	11.0	....	100	....	576	1472	4946	4062	0.8174	0.9472	0.9713	0.9904
			200	....	561	1445	3663	2382	1.0000	1.0000	1.0000	1.0000
			300	....	566	1448	3015	1745	1.0000	1.0000	1.0000	1.0000
	14.0	....	100	....	564	1582	5023	3881	0.8152	0.9635	0.9717	0.9929
			200	....	553	1553	3759	2239	1.0000	1.0000	1.0000	1.0000
			300	....	555	1559	3116	1640	1.0000	1.0000	1.0000	1.0000
	17.0	....	100	....	554	1674	5096	3802	0.8112	0.9725	0.9715	0.9945
			200	....	546	1651	3844	2136	1.0000	1.0000	1.0000	1.0000
			300	....	547	1654	3204	1563	1.0000	1.0000	1.0000	1.0000

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* 100 per cent air means the amount that is just sufficient to allow x and y to become unity if the temperature and pressure permit.

Referring to the notation and the small diagrams at the top of this table, it may be seen from the tabulated values of T_3, x_3 and y_3, that both the Otto and Diesel ideal cycles would have this temperature so high, *when using* 100 *per cent air or less*,[15] that chemical equilibrium would be attained before all the CO and H_2 could be burned. Therefore, in such cases, considerable burning would continue during the expansion while the absolute temperature falls from T_3 to T_4. Even at the end of the expansion all of the fuel might not be completely burned, as is shown by the values of x_4 and y_4, which are slightly less than unity in some cases with 100 per cent air, and naturally very much lower with only 75 per cent air. Clearly, the dissociation may be serious in the ideal Otto cycle, using less than 100 per cent air.

(**h**) **Dissociation in actual Otto and Diesel engines** is appreciably less than in the corresponding ideal, because the maximum temperature is so much lower. This reduction in temperature is chiefly due to the transfer of heat from the working substance to the walls of the combustion chamber in the real engine; but the imperfections of the mixture of air and fuel in the cylinder during the very short time available for the combustion in the high-speed engine also tend to keep the maximum temperature much lower than that calculated for the ideal engine, assuming the same proportion of dilution from the exhaust gases in both cases. An inspection of Fig. 526 shows that, for pressures greater than one atmosphere and temperatures less than 3000 deg. fahr., the amount of dissociation that will occur in the actual engine is very small.

378. The Maximum Temperature Attained During Combustion.—(**a**) Engineers sometimes desire to calculate the maximum temperature that may be expected in the products of combustion of a certain fuel under the different conditions found in furnaces or internal combustion engines. Very often the calculations are made in a crude manner because some of the important factors are completely ignored, due perhaps to lack of knowledge or possibly to the need for a rough result only. On the other hand, when one desires to make this calculation carefully, he must consider a large number of items, some of which involve complications, while the others are relatively simple.

(**b**) The first question to be considered is the magnitude of the *net* or *low heating value of the fuel;* for this is the only one that should enter into the calculation of the temperature attained during combustion. There is never a possibility for the condensation of water vapor until the products of combustion are cooled far below the ignition temperature of the fuel mixture; consequently the latent heat of vaporization of the water vapor is not available to produce a rise in temperature of the products mixture. The use of the net value in such a calculation has nothing whatever to do with the entirely different question of what heating value should be used in determining the thermal efficiency of a heat engine.

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[15] Whether or not it is proper to consider as *ideal* a cycle using less than 100 per cent of air is largely a matter of definition, concerning which considerable information may be found in the paper and its discussion, "Efficiencies of Otto and Diesel Engines," by Ellenwood, Evans, and Chwang, Trans. A.S.M.E., 1928, OGP-50-5-1.

In Table XXXVII (p. 368) the net heating values at constant pressure are given for a temperature of 64 deg. fahr. If the reaction takes place at constant volume, the heating values that should be used for extreme accuracy will be slightly modified. The relation between the low heating value at constant pressure $(L.H.V.)_p$, in B.t.u. per lb. mol, and that at constant volume $(L.H.V.)_v$ is given by Eq. (531) following, which holds rigidly only for *gaseous fuels* and for *gaseous products* of combustion before any vapor may have been cooled sufficiently to approach its point of condensation.

$$(L.H.V.)_p - (L.H.V.)_v = -\ 1.985\ (\Delta n)T \quad . \quad . \quad . \quad (531)$$

In this equation Δn represents the *excess* of the molecular volume of the products mixture over that of the fuel mixture per pound mol of fuel; the constant, 1.985, is obtained by dividing the universal gas constant,[16] 1544, by 778; and T is the absolute temperature, deg. fahr., of the fuel and air before combustion. From this equation it is evident that when the reaction causes a decrease in molecular volume, the heating value at constant pressure will exceed that at constant volume, and vice versa. For example, in the reaction $CO + \frac{1}{2}O_2 = CO_2$, $\Delta n = -\ \frac{1}{2}$, and therefore, if $t = 64$ deg. fahr., $-1.985\Delta nT = \frac{1}{2} \times 1.985 \times 524 = 520$ B.t.u. per lb. mol of CO, or nearly 20 B.t.u. per lb. more for the value at constant pressure than at constant volume. For gasoline vapor (C_8H_{18}) the reaction is $C_8H_{18} + 12.5O_2 = 8CO_2 + 9H_2O$. Hence $\Delta n = +\ 3.5$, and for the same temperature as before, $-\ 1.985\Delta nT = -3640$ B.t.u. per lb. mol of C_8H_{18}; or roughly, 30 B.t.u. per lb. is the amount by which the net heating value of gasoline at constant volume exceeds its value at constant pressure.

(c) In *calculating the maximum temperature attained* during the combustion of any fuel with air the following items should be considered:

(1) The *rate at which heat is developed* by the reaction. This value is dependent chiefly upon the rate of firing, but secondarily upon the

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[16] See page 76, Part I.

heating value of the fuel and the completeness of its combustion within the high-temperature zone.

(2) The *rate of heat transmission* from the gases and vapors within the combustion zone to other bodies, such as the water walls of a boiler furnace or the cylinder walls and piston of an internal combustion motor.

(3) The *rate at which heat is transformed* into mechanical energy during the reaction, as in a Diesel engine during the injection period.

(4) The *weight of the mixture within the combustion chamber* per unit weight of fuel. This depends chiefly upon the *ratio of air to fuel,* but in the case of the internal combustion engine the weight of the residual gases is also an item of considerable importance.

(5) The *instantaneous specific heat of the products of combustion.* This will depend chiefly upon the temperature and composition of the products mixture and only slightly upon the pressure.

(6) The *instantaneous specific heat of fuel mixture* for the temperature range involved between that at entrance to the combustion chamber and ignition. This is dependent largely upon the ratio of air to fuel, but also upon the temperature and relative humidity of the air entering the combustion chamber.

(7) The *temperature of the fuel mixture* entering the combustion chamber; *also its pressure,* if the fuel is a liquid, vapor, or gas.

(8) The *effect of chemical equilibrium* on the limitation of the temperature attainable. This item is negligible in most of the actual combustion processes occuring in engineering applications, on account of the magnitude of item (2); but it has been shown in the previous section to be of importance in some cases in which adiabatic combustion is involved.

Then, if sufficient data are available in any specific case to enable all of the above items to be known, or estimated fairly accurately, the application of the law of conservation of energy, together with the variable specific heat equations, as given in Table E of the Appendix, will enable one to write an energy equation from which the maximum temperature may be calculated. Clearly the accuracy obtainable will be dependent upon one's knowledge pertaining to these eight items, of which (1), (2) and (8) are not easily determined. The complete analysis involving dissociation is complicated and beyond the scope of this text. To those who desire further information regarding such an analysis for adiabatic combustion and also for various percentages of heat transfer, the reader is referred to the Bulletin[17] of Goodenough and Felbeck. Some of their calculated results are given in Table XL for certain gases

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[17] No. 139, University of Illinois, 1924.

TABLE XL

Calculated Explosion Data for Various Gas-Air Mixtures

(10 Per Cent Heat Loss)

No.	Gas $t_1 = 60°$ F. $p_1 = 1$ atm.	Percentage of Air Required for Complete Combustion					
			100	110	120	130	140
1	Hydrogen................	T	4798	4676	4526	4385	4220
		y_0	0.952	0.976	0.987	0.992	0.995
2	Carbon Monoxide.........	T	4732	4668	4582	4488	4382
		x_0	0.850	0.885	0.916	0.939	0.958
3	Carburetted Water Gas *...	T	4632	4528	4388	4240	4097
		x_0	0.842	0.908	0.948	0.970	0.983
		y_0	0.974	0.985	0.991	0.996	0.998
4	Cambridge Coal Gas.......	T	4580	4460	4310	4160	4010
		x_0	0.847	0.917	0.955	0.975	0.988
		y_0	0.974	0.986	0.992	0.995	0.998
5	Natural Gas *.............	T	4516	4398	4240	4080	3920
		x_0	0.864	0.933	0.969	0.981	0.989
		y_0	0.977	0.989	0.994	0.997	1.000
6	Producer Gas *............	T	3858	3780	3674	3575	3470
		x_0	0.969	0.989	0.995	0.998	1.000
		y_0	0.994	0.998	0.999	1.000	1.000
7	Blast Furnace Gas *.......	T	3600	3538	3459	3375	3288
		x_0	0.985	0.996	1.000	1.000	1.000
		y_0	0.997	1.000	1.000	1.000	1.000

Note.—T given in deg. F. (abs.)

x_0 = fraction of CO burned to CO_2 when equilibrium is attained.

y_0 = fraction of hydrogen burned to H_2O when equilibrium is attained.

* For composition of these gases see the following table:—

Gas	Constituents—Percentage by Volume					
	H_2	CO	CH_4	C_2H_4	CO_2	N_2
Carburetted Water Gas..............	22	28	31	13	4	2
Natural Gas........................	2	1	94	......	......	3
Producer Gas.......................	14	20	3	......	9	54
Blast Furnace Gas..................	3	28	......	......	10	59

burned at constant volume with various amounts of air. A 10 per cent loss of heat to the walls of the combustion chamber is assumed to take place during the period of attainment of the maximum temperature; the initial temperature is taken as 60 deg. fahr. and the initial pressure as one atmosphere. This table shows clearly that excess air reduces the maximum absolute temperature T but increases the completeness of the combustion at that temperature, as shown by the higher values of x_0 and y_0, which represent the fractional parts of CO burned to CO_2 and of H_2 burned to H_2O, respectively, when equilibrium is attained due to the high temperature. As soon as the temperature is reduced from any cause the further burning of the fuels would continue.

(**d**) Much **valuable experimental data** *regarding the maximum temperature and the rate of combustion* of various gaseous mixtures under high pressures are given by Bone, Newitt and Townend.[18] Their investigations show that for all of their mixtures the maximum temperature attained during combustion with a fixed initial temperature increases rapidly with higher initial pressures up to about 50 atmospheres after which this rate of increase is much less and becomes very small above 100 atmospheres. The reason for this is not known but those who made the experiments suggest that it is probably due to the reduced heat losses and the greater opacity of the gaseous products to radiation; greater dissociation has also been suggested as a possible reason, but their results indicate that with very high initial pressures the degree of dissociation at the maximum temperature is kept substantially constant by the opposite effects produced by the high pressure and the high temperature.

Another of the many important facts brought out by their experiments is that *water vapor need* **not** *be mixed with carbon monoxide and oxygen in order to have an explosive mixture;* this is contrary to belief previously held. Their experiments show (1) that *the energy required by the minimum igniting spark increases very rapidly as the fraction of water vapor in the mixture is decreased*, (2) that *a sufficiently powerful spark will ignite a strictly dry mixture at atmospheric pressure*, and (3) that *the higher the initial pressure of the dry mixture the easier it is to ignite and the more complete is the combustion.*

Further information regarding the effect of mixture ratio, pressure and temperature on the inflammability of various mixtures is given in the next section.

379. Limits of Inflammability of Gases and Vapors.—(a) An **inflammable mixture** is one in which a flame is propagated from layer to layer of the mixture without the continuation of the source of heat that started the inflammation. If the proportion of the fuel in the mixture is reduced to the minimum that will propagate the flame through the mixture the *lower limit of inflammability* is reached; and conversely, the *higher* or *upper limit* is attained when the mixture is so rich with fuel that further enrichment will prevent flame propagation. The *lower limits of most gases are nearly the same in oxygen as in air, but the higher limits of all inflammable gases are much greater in oxygen than in air.*

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[18] Gaseous Combustion at High Pressures, Longmans, 1929.

(b) **The calculation of the limits of inflammability** of a gas involves so many factors that no one, according to Coward and Jones,[19] has yet been able to do this from the fundamental physiochemical data. On the other hand, for a known mixture of gases the upper and lower limits of inflammability may be calculated easily, and to a fair degree of accuracy generally, if the limits in air of each constituent are known from experimental data. This is done by the **mixture law of Le Chatelier** applied to the inflammability of gases. Thus if f_1, f_2, and f_3 are the respective fractions, by volume, of each constituent in the original mixture free from air and inert gases, so that $f_1 + f_2 + f_3 = 1.00$, and if L_1, L_2 and L_3 are the respective limits in air (either lower or higher) of these constituents, then the corresponding limit of the mixture is

$$L = \frac{1}{f_1/L_1 + f_2/L_2 + f_3/L_3} \quad . \quad . \quad . \quad . \quad . \quad (532)$$

This law has been found to hold best for mixtures of carbon monoxide, hydrogen and methane either in pairs or three together; it also holds well for water gas, coal gas and natural gas; but it is only approximately true for mixtures of hydrogen-ethylene-air, acetylene-hydrogen-air, hydrogen-sulphide-methane-air, hydrogen-sulphide-hydrogen-air, and mixtures containing carbon disulphide.

(c) **The most important factors,** aside from the composition of the mixture, affecting the inflammability of gases and vapors are the *temperature* and *pressure* of the mixture, the *diameter* and *length* of the *container*, the *direction of flame propagation*, and turbulence.

In making experiments the container should be of sufficient size that the walls produce only a negligible cooling effect; in large vessels the convection currents affect the flame travel so that near the limit mixtures the flame will travel successfully upward but not downward.

Weak *sources of ignition* are not satisfactory when trying to fire mixtures near the limits of inflammability, because such mixtures may be very hard *to ignite*; but this should not convince one that the limits of inflammability of a mixture are dependent on the strength and nature of the ignition. The determination of the *limits of ignitibility* of an inflammable mixture by different sources of ignition is not the same as the determination of the limits of inflammability as defined in Sect. **379** (**a**). Whether the source of ignition should be considered in the list of factors affecting inflammability seems to be debatable. Thus, on page 2 of the Bureau of Mines, Bulletin 279, Coward and Jones state that " it is axiomatic that the limits are unaffected by variations in the nature and strength of the source of ignition." Also one of these authors states [20] that: "A very intense source of ignition introduced into a mixture incapable of propagating flame will inflame more mixture than a weaker source. Aside from this, the source of ignition has no effect on the observations when the tests are carried out in an apparatus of the proper dimensions. The true test of an inflammable mixture is whether or not it continues to propagate flame after the influence of the source of ignition has been dissipated."

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[19] See Limits of Inflammability of Gases and Vapors, Bulletin 279, Bureau of Mines, 1931, p. 12.

On the other hand reference has already been made, in Sect. **378** (**d**), to the work of Bone, Newitt, and Townend in igniting and burning dry CO when a powerful spark was used. Furthermore, Wendlandt [21] states that " the spread of the ordinary flame through an explosive mixture of gases depends in every respect on the influence of the experimental conditions imposed upon it. All data of this nature really rest on the kind of ignition, its position, the form and material of the container and on the way the reaction runs its course; in short, upon the conditions imposed upon the reaction by the experimental method employed, as temperature and pressures."

(**d**) For **pressures less than atmospheric** the range of inflammability of most combustible mixtures is reduced as the pressure becomes lower. When the pressure of the mixture is reduced only slightly below the normal atmosphere, the effect on both the upper and lower limits is hardly noticeable; but when the pressure becomes less than about half an atmosphere, the range of inflammability decreases rapidly as the pressure is further reduced; and all experiments indicate that if the pressure becomes sufficiently low, the two limits would coincide, or, in other words, no mixture would propagate flame at a very low pressure. The exact values of these low-pressure limits for various fuel mixtures have not been experimentally determined. However, all results show that the two-limit curves approach each other as the pressure is decreased, and they would undoubtedly meet at some low pressure. This general truth may be illustrated somewhat imperfectly with Otto engines operated with various mixtures and different values of the inlet manifold vacuum. The limiting vacuum at which the engine will fire with any mixture will generally be found to be between 18 and 24 in. of Hg referred to a 30-in. barometer, the exact value depending upon the particular engine and fuel used. Various engines yield different results chiefly because the proportion of diluent in the mixture due to the residue in the clearance volume is different. This factor is mainly dependent upon the clearance, valve timing, and exhaust pressure; but the design of the intake manifold, exhaust manifold, and cylinder head, the location of the spark plug, the intensity of the ignition, and the temperature of the walls of the combustion chamber are also important.

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[20] See page 2, Technical Paper 450, Bureau of Mines, Inflammability of Mixed Gases, by G. W. Jones, 1929.

[21] "Experimental Investigations Concerning the Limits of Detonation in Mixtures of Explosive Gases," Z. f. Physic. Chem. 110, 673. Translated in Tech. Memorandum, No. 553, 1930, N.A.C.A., page 1.

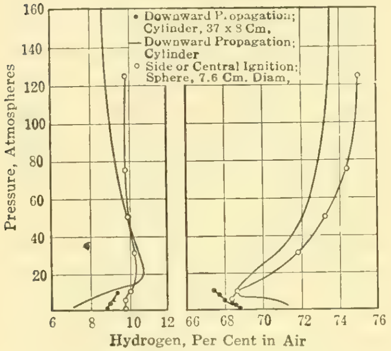

Fig. 527.—Effect of High Pressures on the Inflammability of Hydrogen in Air.

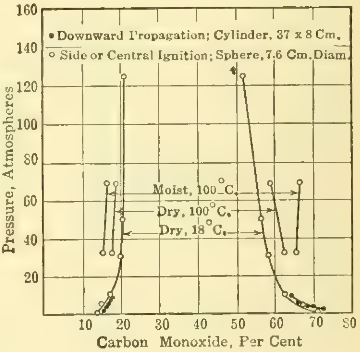

Fig. 528.—Effect of High Pressures on the Inflammability of Carbon Monoxide in Air.

(e) For **pressures greater than atmospheric** an increase in the pressure of the mixture has been found to increase the range of inflammability of some mixtures and to decrease the range of others. Two illustrations of these facts are shown by the curves in Figs. 527 and 528, as given by Coward and Jones [22] for hydrogen and carbon monoxide in air. These curves give the results of several different observers and on the whole are in very good agreement considering the experimental difficulties involved. Clearly, the range of inflammability of hydrogen is at first narrowed and then widened by an increase of pressure above atmospheric; but for dry carbon monoxide the range is narrowed by the high pressures. The range for moist carbon monoxide is apparently not much affected by the increased pressure.

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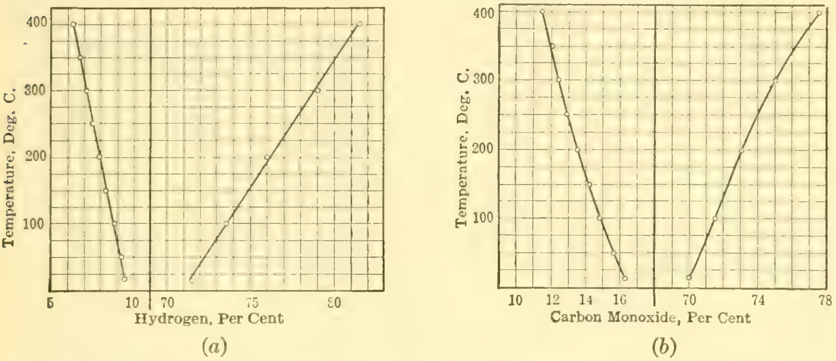
[22] Bulletin 279, Bureau of Mines.

(f) **Increasing the temperature of a mixture** maintained at constant pressure increases the range of inflammability almost in direct proportion to the increase in temperature. Illustrations of this are shown by the curves in Fig. 529 (*a*) and (*b*), as given by Coward and Jones for hydrogen and carbon monoxide in air. Their curves for ammonia and air are given in Fig. 530, which also shows the effect of the direction of flame propagation on the range of inflammability.



(*a*) (*b*)

FIG. 529.—Effect of Temperature on the Range of Inflammability of Hydrogen and Carbon Monoxide in Air.

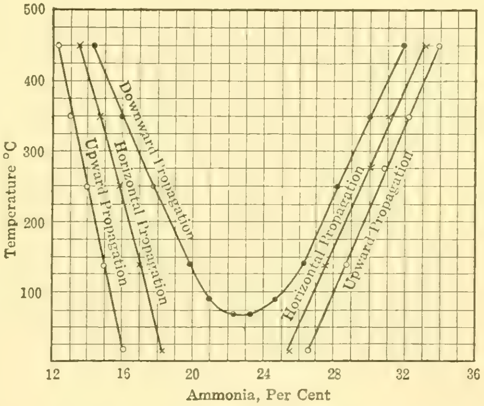


FIG. 530.—Effect of Temperature on the Range of Inflammability of Ammonia in Air.

(g) **Approximate values of the lower and higher limits of inflammability of single gases and vapors in air** at ordinary temperatures and pressures are given in Table XLI as compiled from different sources by Coward and Jones. Their values for some important industrial mixtures are also given in Table XLII. Both tables give the limits for *upward propagation* of the flame because such values are wider than those for horizontal or downward propagation; hence they represent safer limits to use in industrial operations involving inflammable mixtures.

TABLE XLI

Approximate Limits of Inflammability of Single Gases and Vapors in Air at Ordinary Temperatures and Pressures

Gas or Vapor	Lower Limit, Per Cent by Volume	Higher Limit, Per Cent by Volume	Gas or Vapor	Lower Limit, Per Cent by Volume	Higher Limit, Per Cent by Volume
Hydrogen	4.1	74	Ethylene oxide	3.0	80
Ammonia	*16*	*27*	Acetaldehyde	*4*	*57*
Hydrogen sulphide	*4.3*	*46*	Furfural (125° C.)	2	......
Carbon disulphide	*1.0*	*50*	Acetone	3	*11*
Carbon monoxide	12.5	74	Acetone (turbulent mixture)	*2.5*	......
Methane	5.3	14.0	Methyl ethyl ketone	*2*	*12*
Methane (turbulent mixture)	5.0	15	Methyl formate	6	20
Ethane	3.2	12.5	Ethyl formate	*3.5*	*16.5*
Propane	2.4	9.5	Methyl acetate	*4.1*	*14*
Butane	1.9	8.5	Ethyl acetate	2.5	*11.5*
Pentane	1.45	7.5	Propyl acetate	2.0	......
Ethylene	3.0	20	Butyl acetate (30° C.)	1.7	......
Acetylene	3.0	......	Ethyl nitrite	*3*	......
Acetylene (turbulent mixture)	2.3	......	Methyl chloride	8	19
Benzene	1.4	7	Methyl bromide	13.5	14.5
Toluene	1.4	*7*	Ethyl chloride	4	15
Cyclohexane	1.3	8.3	Ethyl bromide	7	11
Methyl cyclohexane	1.2	......	Ethylene dichloride	6	*16* *
Methyl alcohol	7	......	Dichlorethylene	10	13
Ethyl alcohol	4	*19*	Vinyl chloride	4	22
Ethyl ether	1.7	26	Pyridine (70° C.)	*1.8*	*12.5*

* 125° C. Values in italics are approximate.

TABLE XLII

Approximate Limits of Inflammability of Some Industrial Mixtures of Gases and Vapors in Air at Ordinary Temperatures and Pressures

Gas or Vapor	Lower Limit, Per Cent by Volume	Higher Limit, Per Cent by Volume	Gas or Vapor	Lower Limit, Per Cent by Volume	Higher Limit, Per Cent by Volume
Benzene	1.1	......	Natural gas	4.8	13.5
Gasoline	1.4	6	Illuminating gas	5.3	31
Water gas	6 to 9	55 to 70	Blast-furnace gas	*35*	74

380. Rates of Reaction, Explosive Waves.—(a) The **reaction of fuel and oxygen** may take place so slowly below the ignition temperature as to be scarcely noticeable, and yet, after a long time, the resultant products are found to be identical with those obtained from rapid combustion; the magnitude of the energy transformations involved are also the same in both cases, but these two widely different rates of reactions produce very distinct external effects, such, for example, as radiation and sound. A somewhat similar condition exists when the rate of reaction reaches a still higher value than that called "normal burning." Thus if the reaction becomes sufficiently rapid an "**explosive wave**" or a "**detonation wave**" may develop. Here again an abrupt change seems to be involved between these two cases in so far as one may judge from the outward manifestations. Just how these rates of reactions are related one to the other, and how the components of the gas against the wave front are brought to a condition of activation so very rapidly, as in some forms of detonation, cannot be fully explained at the present time by any one; but much effort has been expended over a period of years by various investigators in trying to ascertain facts and to formulate theories that will agree with the observations obtained from different combustible mixtures in various types of combustion chambers and subject to wide ranges of pressure and temperature. Only a few of these investigations can be even mentioned here, but sufficient references will be given to enable one to begin a comprehensive study of the subject, which is of the utmost importance to engineers because of its direct bearing on the development of automotive engines.

(b) The **normal velocity of flame propagation** within a cool gas is very small, less than 10 ft. per sec. generally, depending upon the mixture. The measurement of flame propagation involves many experimental difficulties; and the value of a considerable portion of the available data is appreciably reduced because a large number of experimenters, working with combustion at constant volume, have failed to distinguish between the absolute and relative movement of the reaction zone. The *velocity relative to the unburned gases* is the quantity desired, but the movement of the unburned gases while being compressed and pushed away from their former positions by the expansion of the gases within the reaction zone in a constant volume container is difficult to measure; and the effect of pressure on the reaction rate and on the velocity of flame propagation is impossible to determine except by a combustion chamber in which constant pressure may be maintained.

In 1889 Michelson made some extensive and careful studies regarding the rate of propagation of the normal flame at constant pressure; he obtained a smooth curve when velocities were plotted against volumetric mixture ratios. This has led to the very recent and valuable work of Stevens,[23] who has devised a novel method of igniting a mixture held temporarily inside a soap bubble that is surrounded by an inert gas that may be maintained at any desired pressure within a strong container. With this device he has obtained some very accurate results for a large number of explosive gases. He has also developed a *statistical relation between the linear rate of propagation of the reaction zone within the gases and their molecular concentrations,* and then checked this relation by means of his experimental results. Stevens found that the check was very satisfactory in all cases for which the stoichiometric equation could be written for the completed reaction. One example of his work is shown in Fig. 531, which gives the results for mixtures of carbon monoxide and oxygen under various pressures. The abscissas are the *initial partial pressure* proportions of the reacting components, and the ordinates are the velocities of the flame relative to the active gases.

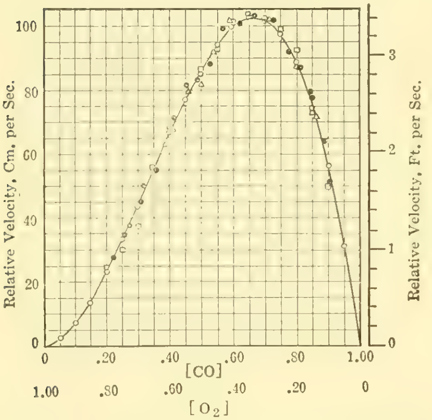


FIG. 531.—Relation of the Normal Rates of Propagation of the Reaction Zone and the Concentration.

Initial Pressures, mm. of Hg. abs.: △ = 200, ● = 760, □ = 2530, ○ = calculated points.

The experimental and calculated values are seen to be in almost perfect agreement and to show clearly, for the low-pressure range involved, that the *rate of propagation of the reaction zone measured relative to the active gases is independent of pressure.* Stevens has studied a very large number of reactions and the results are similar to those given for carbon monoxide, although the experimental and calculated values are not always in such remarkable agreement. He also points out that the *rate of energy transformation within the reaction zone is proportional to pressure* although the propagation of the normal combustion wave is independent of it. Under some conditions combustion of a gaseous mixture may not proceed under normal conditions.

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[23] "The Gaseous Explosive Reaction—The Effect of Pressure on the Rate of Propagation of the Reaction Zone and upon the Rate of Molecular Transformation," F. W. Stevens, National Advisory Committee for Aeronautics Report 372, 1930; also The Scientific Monthly, June, 1931.

(c) The **explosive wave** that may occur in gaseous combustion was first observed and measured in 1881 by Berthelot and Vieille but also nearly simultaneously by Mallard and Le Chatelier; it is often spoken of as the *Berthelot explosive wave,* which is a form of reaction characterized by an unusually high velocity or speed of propagation. For example, Dixon[24] observed the "normal explosive wave" of various fuel mixtures to have velocities, in meters per second as follows: 3530 for $H_2 + O + 5H$, 1707 for $H_2 + O + 5O$, 1822 for $H_2 + O + 5N$, 2528 for $CH_4 + O_2$ and 2166 for $CH_4 + 4O_2$. These are called "*normal explosive waves,*" or "*normal detonation waves,*" because each mixture gives a detonation velocity that remains constant when the wave is transmitted through a long tube filled with the given mixture. Each explosive fuel mixture has been found to have *two characteristic normal rates of flame propagation* relative to the active gases, one for *normal burning* and the other for *normal detonation.* Under certain conditions, however, *detonation waves of very much higher velocity than normal* have been found to exist for a short time. When any detonation wave strikes the cylinder head or piston of an engine, a sharp metallic sound or "knock" results. (A discussion of complex gas-air mixtures giving maximum uniform flame speeds is given in Sect. **435.**)

381. Detonation in the Internal Combustion Engine.

—(a) The *complete explanation of detonation* that is so common in internal combustion engines probably cannot be given by any one yet, although constant study has been given to the subject for many years and much valuable information has been acquired.

To assist in visualizing the probable action within an engine having combustion at approximately constant volume, consider a spherical container filled with a combustible mixture and ignited at the center. At first only a very small portion of the mixture near the source of ignition constitutes the zone of the reaction, but as this zone spreads in all directions at the normal rate of propagation, the area of the wave front and the pressure within the sphere both increase enormously. The rate of energy liberation is directly proportional to the area of the wave front. This rate of liberation is also increased by the higher pressure which increases the density of the unburned mixture. Consequently the temperature of the unburned gas is raised rapidly, chiefly by its compression due to the expansion of the reaction zone, but also slightly by the transmission of heat from this zone. Hence, after a very short interval of time, the temperature and density of the unburned mixture are increased rapidly. The high temperature and density produced in an engine during normal combustion very soon after ignition *favor the development of an explosive wave,* probably starting at the flame front and passing at an enormous velocity through the remainder of the unburned mixture, but especially through the hottest parts of it. Consequently, the portion of the mixture that is furthest from the source of ignition and therefore the last to burn under normal flame propagation is the part through which the explosive wave will spread unless this part is kept relatively cool.

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[24] See N.A.C.A. Tech. Memo. 506, 1929, Impact Waves and Detonation, by R. Becker, Tr. from Z. für Physik, Volume VIII.

When an explosive wave has once developed in a mixture, its rate of propagation is so much greater than that of sound that the pressure in the unburned mixture ahead of this wave is not changed, since pressures can not be transmitted through a gas at velocities greater than that of sound in the given medium. The period of waiting for the explosive wave to reach a certain point is extremely short since the wave is traveling at such an enormous velocity,[25] say 6000 ft. per sec. or more; this would mean that such a wave could travel clear across a 3-inch cylinder in $\frac{1}{24000}$ sec.

(**b**) The **photographic studies of the flame** inside an engine cylinder, as recently made by Withrow and Boyd,[26] seem to offer one of the best means of studying the complicated phenomenon of combustion in an engine. Their studies were made on an engine equipped with an ingenious arrangement of a quartz window and a camera containing a moving film. Their conclusions are as follows:

1. Starting at the spark plug a flame, or narrow combustion zone, moves progressively through the charge. Oxidation of the fuel is apparently complete within this

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[25] Investigations are not in agreement as to the exact manner in which a detonation wave spreads so rapidly. For example, Mallard and Le Chatelier believed that ignition in such a wave front was brought about by the high temperature resulting from excessively high compression. On the other hand, Becker (N.A.C.A. Tech. Memo. 506, p. 22) states: "It seems necessary to conclude that in the case of detonation the heating of the particles that have reached the wave front, to their ignition temperature, takes place principally through heat conduction from the particles last burned (that is, by the impacts of individual molecules from the very thin wave of high temperature). The most successful experimenters in the field of gas detonation, Berthelot and Dixon, also made use of such an assumption. They conceived that from the hot wave front, molecules were thrown forward that shared their high kinetic energy by impact with the molecules in front of the wave and thereby brought them to a condition of activation."

Wendlandt (N.A.C.A. Tech. Memo. 554, p. 11), however, believes that "the solution will not depart far from the views set forth by Le Chatelier." Thus this question still remains very much of a puzzle to all who have studied it.

[26] "Photographic Flame Studies in the Gasoline Engine" by Lloyd Withrow and T. A. Boyd, Industrial and Engineering Chemistry, May, 1931.

narrow zone of combustion, but the products of combustion to the rear of the flame front continue to emit light for some time.

2. During normal or non-knocking combustion, the time required for the average flame to travel across the combustion space (under the conditions of these tests) was about 40 degrees of crankshaft revolution.

3. A knocking explosion differs from a non-knocking one only in the way the last portion of the charge burns. The difference is this: Whereas in a non-knocking explosion the flame continues to move at a comparatively constant velocity clear to the end of the combustion space, in a knocking explosion the latter portion of the charge inflames at a much higher rate than normal. This rate is often so high at the instant of knock the flame appears simultaneously throughout the whole of the portion of the charge still remaining to be burned.

4. The extremely high rate of inflammation in that portion of the charge which burns at the instant of knock is apparently due to auto-ignition occurring within it. This may be caused by temperature induced within that part of the charge by adiabatic compression.

5. The violence of the knock is determined by how large a portion of the total charge is involved in the spontaneous inflammation, or the amount of it still remaining to be burned at the instant knock occurs.

6. The very rapid, and often substantially instantaneous, inflammation that occurs within the portion of the charge which burns at the instant of knock is accompanied simultaneously by a very rapid rise in cylinder pressure. The magnitude of this pressure rise increases along with the intensity of the knock.

7. The one effect upon combustion of the presence of lead tetraethyl in the gasoline is to prevent the extremely rapid inflammation of the latter portion of the charge, and the accompanying pressure rise, which is the knock. Lead tetraethyl has no effect upon the velocity or the character of the flame prior to the time at which knock would have occurred in its absence.

From these conclusions, as well as from the discussion in the previous section, one would expect that the degree of detonation produced in an engine would be determined by a number of factors that will now be considered.

(c) The **compression ratio** of an engine is a factor of the utmost importance regarding detonation because the temperature and pressure of the mixture and also the proportion of the diluent are so dependent upon this ratio. High temperature, high pressure, and a large proportion of fresh charge all promote rapid rates of reaction. The proportion of the diluent, or exhaust gases, present during combustion is very much less in engines with small clearances, or high-compression ratios, than in those with low-compression ratios because the volume of this gas is directly dependent upon the clearance, whereas its density is not greatly affected thereby.

When a high-compression engine is *throttled appreciably* the weight of fresh charge is greatly reduced but the weight of exhaust gases remaining in the cylinder is not materially affected; thus the proportion of diluent is greatly increased by throttling, which also reduces the pressures prevailing during the combustion period, and both of these factors reduce the rate of the reaction and thus the maximum temperature of the flame. Furthermore, with the engine throttled, the exhaust valve, piston and walls do not become so hot; and thus all of these factors combine to make it possible to avoid detonation by throttling an engine that would otherwise " knock " or " pink " badly with a given fuel.

(d) The **time of ignition** in a high-compression motor has an important bearing on the amount of detonation produced. An illustration of this relation is given by the curves in Fig. 532, as obtained by Zuck.[27] His tests were made upon a single cylinder 5 × 7-inch N.A.C.A. Universal test engine,[28] which permitted a wide variation in compression ratio, even when the engine was running.

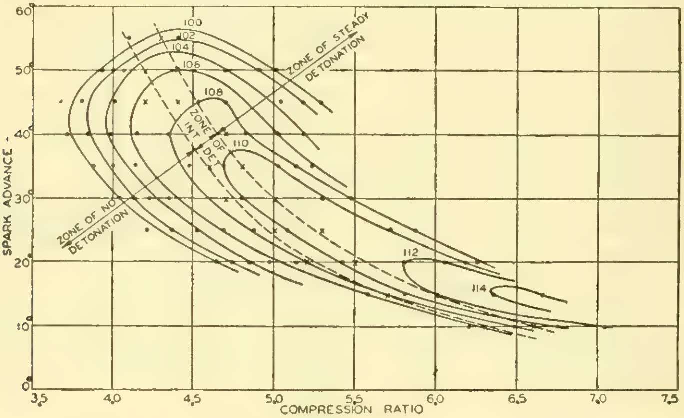

FIG. 532.—Relation of Compression Ratio, Spark Advance, and Detonation for Constant Values of B.M.E.P. Avg. mixture ratio, 14.27 to 1; R.P.M. = 1000; full throttle. The numbers on curves are b.m.e.p.s. in lb. per sq. in.

The dotted lines represent the boundaries between the respective zones of no detonation, intermittent detonation,[29] and steady detonation as determined by ear. Note that these dotted lines run almost parallel with the full curves of b.m.e.p. over quite a range of compression ratios; but particularly true is this of the range in which the maximum power is obtained. Observe that, with a given compression ratio, the same power may be obtained without detonation by having a retarded spark as with an advanced spark and excessive detonation, and that the maximum power is near the border line of steady detonation for each compression ratio.[30]

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[27] "Engine Performance at High Compression Ratios," by H. E. Zuck, Eng. Research Circular No. 6, Univ. of Mich., 1931.

[28] For complete description see N.A.C.A. Tech. Report No. 250.

[29] The zone of intermittent detonation for these curves is defined as beginning "at that point where the detonation which is first distinguishable as a low rumble begins to include occasional sharp knocks."

(e) The **size of the cylinder** of an internal combustion engine is also a factor in producing detonation. From the preceding analysis one would expect large cylinders to produce more severe detonation than small ones having the same compression pressure and temperature of the charge. This is confirmed by experience with automotive engines; but in the actual engine the *effect of the cooling area in contact with the last part of the gas to be burned is also a factor of the utmost importance,* and in the small cylinder the ratio of this area to volume is greater than in a larger one having the same type of combustion chamber. Furthermore, a small cylinder will require a small exhaust valve, which may be kept cooler than a larger one, and thus there is less danger of overheating the adjoining mixture. Also the temperature of the head of a light piston may be kept cooler in the small bore cylinder than in the larger one. These four factors all tend to produce less detonation in the small cylinder than in the large one having the same compression ratio, shape of combustion chamber, location of spark plug, jacket water temperature, fuel mixture, engine speed, load, and spark timing.

(f) The **shape of the combustion chamber** in an automotive engine having a high-compression ratio has a most important effect on detonation, as might be expected from the preceding discussion. This chamber is made to assume a rather large variety of forms in modern engines; but in all cases the designer is *aiming to prevent detonation and to secure smooth* operation of the engine. The question of what shape is best, all factors considered, has become one of intense interest and discussion among automotive engineers; and naturally the most prominent designers in this field are not in complete agreement regarding all phases of the complicated problem. However, the two following points seem to be generally accepted at the present time as the most important ones regarding the *best shape* to reduce detonation: (1) *the last gas to be burned should be well cooled,* as may be done in a shallow space of about $\frac{1}{8}$ to $\frac{1}{4}$-in. thickness directly above the piston; (2) The *spark plug should be located somewhat near the exhaust valve,* since this is the region of the combustion chamber having the least cooling effect on the mixture. This last statement applies particularly to the poppet-valve engine, since there is no hot exhaust valve in the sleeve valve type.

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[30] Additional information pertaining to this general subject may be found in Tech. Report No. 272, of the N. A. C. A., 1927; "The Relative Performance Obtained with Several Methods of Control of an Overcompressed Engine Using Gasoline," by Gardner and Whedon.

If the shape of the combustion chamber is not properly worked out when trying to overcome detonation, the combustion may be so rapid that the rise of pressure in the cylinder becomes so sudden that the engine operates with a noticeable roughness that is very undersirable. *Smoothness of operation* can probably best be attained by (*a*) restricting the initial volume burned, (*b*) providing the maximum area of flame spread at intermediate inflammation and (*c*) restricting the flame spread at the end of the burning in the main chamber. These items regarding smoothness are presented as summarized by Taub [31] in an excellent article covering a general review of the work of Janeway, Ricardo and Whatmough, who have been very prominent in the development of the combustion chamber used in the modern engine.

(g) The **fuel used** in an engine is of the utmost importance when considering detonation. Mention has already been made (page 351) of certain fuels and compounds that tend to reduce detonation, and of other fuels that are very bad in this respect. The relative volumes, as given by Whatmough,[32] of the most common anti-knock substances that may be mixed with a fuel to prevent detonation are:

Anti-knock Substance	Per Cent by Volume
Benzol	25 to 40
Alcohol	12 to 20
Aniline	3 to 5
Iron carbonyl	0.2
Lead tetraethyl	0.08

In comparing the effect of different substances in reducing detonation, some form of instrument became necessary to obtain reliable data: the Midgeley bouncing-pin indicator,[33] which includes a modified form of Dickinson's bouncing pin, has been very satisfactory in this work.

The question of why minute quantities of some substances are so effective in reducing detonation has been studied by various scientists and engineers; but no complete theory of this action during combustion has been generally accepted. Thus Clark and Henne [34] state that " the mechanism of knock inducers and suppressers is far more complicated than can be adequately accounted for by the nine or more theories that have been proposed." However, the work of these men indicates that spectroscopy affords a possible key to the mystery, as higher frequency radiations are produced when detonation occurs than with normal combustion.

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[31] "Combustion Chamber Progress Correlated," Alex Taub, S.A.E. Journal, Oct., 1930, pp. 413 to 443.

[32] See Automobile Engineer, May, 1927, p. 170.

[33] See S.A.E. Journal, Jan., 1922, p. 8.

[34] "Ultraviolet Spectroscopy of Engine-Fuel Flames." S.A.E. Journal, Feb., 1927.

(h) **Preignition** is a term often used in connection with the Otto engine in somewhat different senses by various authors. Basically, the word means *premature ignition* of the mixture in the cylinder. Consequently, the premature ignition of the charge from *any cause* would constitute preignition. The most common causes are: hot electrodes, overheated exhaust valve or piston, incandescent carbon, and too early firing of the spark. High-compression ratio, high temperature of the jacket water and of the intake air all combine to promote preignition. Some authors prefer to limit the term preignition to too early ignition produced from any cause other than the passing of the spark, but this seems to be a restriction of doubtful value. *Preignition* and *detonation* should not be confused, although both produce serious knocks in the engine and are very undesirable; detonation occurs *after* combustion has proceeded to a considerable degree, whereas preignition *starts* combustion prematurely. If the engine is operated under conditions favorable to the development of high temperature in the charge near the end of the compression, both are more likely to occur.

When used in connection with the Diesel, *preignition* implies premature injection of the fuel so that the pressure rises abruptly before the top dead center and thus produces a reduction in the output of the engine, and generally a knock.

(i) **Additional references** are given here to supplement those already mentioned in Sects. **379** to **381** inclusive. The subject of combustion in boiler furnaces is treated in Chapters XXXI to XXXIV inclusive: that in oil engines, in Chapter XXXV.

REFERENCES

Berthelot et Vieille—Bul. Soc. Chim., **40**, 2 (1883).

Bone, W. A., and Townend, D. T. A.—Flame and Combustion in Gases. Longmans, Green and Co.

Clark, G. L. and Thee, W. C.—Present Status of the Facts and Theories of Detonation. Industrial and Engineering Chemistry, Dec., 1925.

Duchene, M. R.—Contribution to the Study of Normal Burning in Gaseous Carbureted Mixtures. N.A.C.A. Technical Memorandums Nos. 547, 548 (1930).

Ellis, C.—Hydrogenation of Organic Substances. D. Van Nostrand Co.

Gouy—Ann. de Chim. et Phys. **5**, 18 (1879).

Haslam, R. T., and Russell, R. P.—Fuels and Their Combustion. McGraw-Hill Book Co., New York.

Janeway, R. N.—Combustion Control by Cylinder-Head Design. S.A.E. Journal, May, 1929.

Jourquet, M. E.—Journ. Math., 1, 347 (1905); 2, 5 (1906).

McCormick, G. R., Upton, G. B., Davis, A. C. and Diederichs, H.—Flame Propagation in Closed Cylinders, Bul. Cornell Univ. Engineering. Exp. Station, Ithaca, N. Y., No. 8 (1927).

Michelson—Ann. d. Physik. u. Chemie, 37, 1 (1889).

Midgley, T. Jr. and Janeway, R. N.—Laws Governing Gaseous Detonation. S.A.E. Journal, Apr., 1923.

Nernst, W.—Physico-Chemical Considerations Concerning the Process of Combustion in Gas Engines. Springer, Berlin (1905).

Ricardo, H. R.—Cylinder Head Design—The Automobile Engineer, July, 1929.

Stevens, F. W.—The Gaseous Explosive Reaction—A Study of the Kinetics of Composite Fuels. N.A.C.A. Technical Report No. 305 (1929).

—— The Gaseous Explosive Reaction at Constant Pressure—The Reaction Order and Reaction Rate. N.A.C.A. Technical Report No. 337 (1929).

—— The Gaseous Explosive Reaction—The Effect of Inert Gas. N.A.C.A. Technical Report No. 280 (1927).

Whatmough, W. A.—Combustion Chamber Design in Theory and Practice. S.A.E. Journal, Sept., 1929.

Wills, F., and others.—Gas Engineers' Handbook. McGraw-Hill Book Co., New York.

Withrow, Lloyd and Rassweiler, Gerald M.—Spectroscopic Studies of Engine Combustion. Industrial and Engineering Chemistry, July, 1931.

Withrow, L., Lowell, W. G., and Boyd, T. A.: Following Combustion in the Gasoline Engine by Chemical Means. Industrial and Engineering Chemistry, Sept., 1930.

CHAPTER XXIX

THE ANALYSIS OF THE PRODUCTS OF COMBUSTION AND ITS INTERPRETATION

382. Introduction.—(a) A quantitative analysis of the exit gases leaving an internal combustion engine, a boiler furnace, or an industrial furnace may be used directly, or indirectly, to derive much valuable information regarding the composition of the fuel burned, the character of the combustion process, the amount of air supplied, the completeness with which the chemical reactions occur, the weight of gases discharged, the probable maximum combustion temperature, and the quantity of energy wasted with the discharge gases; furthermore, gas analyses taken at successive points along the path of the gas flow through boiler passes and flues may be used to determine the magnitude of the air infiltration occurring between these sampling points.

In some of the investigations, in addition to the analysis of the exit gases, it is also necessary to have the ultimate analysis of the fuel. If only the proximate analysis is available, the approximate ultimate analysis can be derived from it by the method given in Sect. **350.**

Fortunately there are available to the engineer very easy methods for making the simple analyses of the exit gases that he ordinarily needs. Often automatic gas-sampling and analyzing instruments are used to give a continuous indication or record regarding the more important components in the gases leaving furnaces or other fuel-burning equipment. Such information serves as a guide in the regulation of the combustion process, or in analyzing the conditions so that the best combustion conditions may be maintained.

(b) The term **products of combustion** is commonly and loosely applied to the mixture of gases and vapors present after combustion, including the inert gases of the original fuel and those resulting from the reactions, and also any combustible gases that have not been burnt. Most fuels contain the combustibles carbon, hydrogen, and sulphur, the products of complete combustion of which, with just the right amount of oxygen, are CO_2, H_2O, and SO_2. For supporting combustion processes in most engineering work the oxygen is supplied from the air rather than in the pure state, and often the amount of this air is in excess of that chemically required for complete combustion. Hence the nitrogen from the air and any excess oxygen act as **diluents,** and are found mixed with the gaseous products of the reaction, together with any inert gases from the fuel. If combustion of the fuel is *incomplete,* the products may also include CO, unburned H_2, gaseous hydrocarbons, and soot, the latter appearing as smoke. When a solid fuel, such as coal, is burned, there may be a residue of mineral ash and some unconsumed combustible. Part of this solid matter may pass off as dust with the gases, while the remainder will be deposited in the ash pit or hopper. In the following discussion, the products of combustion will be assumed to consist only of the gaseous and volatile matter discharged.

As before, the term *exit gases* will be applied broadly to the products, whereas the term *flue gases* will be limited to the discharge from furnaces, and the term *exhaust gases* to the discharge from internal combustion engines.

383. Analysis of Exit Gases.—(**a**) *The determination of the CO_2, CO, O_2 and N_2 content of an exit gas is all that is required of a gas analysis for most engineering purposes;* and often the CO_2 content is all that is needed.

The **method of collecting the gas sample** to be analyzed is as important as the analysis proper; for if the sample is not representative of the entire gas flow at the sampling point, accuracy in the analysis itself is of little value. Choice of the proper sampling point depends upon the purpose of the analysis. In general, pockets back of dampers and at bends and sections of sluggish flow should be avoided.[1] If the combustion process alone is to be studied, the gas sample should be taken at the place where combustion is judged to be complete. If the heat-losses in a boiler furnace are to be determined, the gas sample should be extracted only after there is no possibility of further transfer of heat from the products of combustion to the heating surface. In a multicylinder internal combustion engine, the sample is commonly taken between the exhaust manifold and the muffler, if the engine as a whole is to be studied. If, however, the distribution of the air-fuel mixture to the different cylinders is being investigated through an exhaust gas analysis, the exhaust of each cylinder must be sampled separately.

If the sample is collected over water, the latter should have been previously saturated with the exit gas; otherwise there may result an extremely large error, particularly so in the CO_2 determination, because this gas is very soluble in water.

(**b**) The **apparatus** used by engineers for making the analysis is generally of the **Orsat type,** which determines directly the fractions by volume of CO_2, CO, and O_2 in the exit gas. Either a three- or four-tube Orsat apparatus is generally employed. With this apparatus the gas is drawn successively, first through a tube containing a solution of caustic soda or potash which absorbs the CO_2; secondly, through a tube containing either an alkaline solution of pyrogall or else phosphorus sticks to remove the O_2; and finally, through either one or two tubes of acid cuprous chloride to remove the CO.

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[1] Bureau of Mines Bulletins 12 and 97.

After the absorption of each gas, the remaining sample is brought to atmospheric pressure and the reduction in volume observed in a measuring burette. Assuming constant temperature and pressure, and no evolution or absorption of gas from or by the displacement liquid, this reduction in volume divided by the original volume gives the fraction, by volume, of that gas in the sample. The N_2 content is then computed to balance the volume analysis. *The analysis must be carried on in the order indicated, because alkaline pyrogallate will also remove the* CO_2*, and acid cuprous chloride absorbs the* O_2*.*

Water vapor is always carried by the flue gas; but the collection of the gas over water and the later contact with water in aqueous solutions in the apparatus will probably cause the gas sample to become saturated during the analysis. If this is the case, when a certain volume of gas is removed by a reagent, a proportionate volume of water vapor will also condense. *Thus, the results of the Orsat analysis are exactly the same as though a* **dry** *gas had been analyzed.* If the water-vapor content is to be determined, it is sometimes measured by using mercury as the collecting and displacement liquid.

Any SO_2 present in the flue gas will be absorbed with the CO_2 in the Orsat apparatus. With fuels high in sulphur and low in carbon, as with certain middle western fuels, this may lead to an appreciable error.

(**c**) Since unburned combustible gases appear only in very small amounts, if at all, in the exit gases of *furnaces* which are properly operated, the analysis of these gases by means of the usual Orsat apparatus gives results sufficiently exact for ordinary purposes. With *internal combustion engines,* however, the combustion is generally less complete than that in furnaces, and the exhaust gas may therefore contain not only very appreciable amounts of CO, for which the ordinary Orsat apparatus is unsatisfactory, but also considerable amounts of unburned H_2 and hydrocarbons which do not appear in the ordinary analysis. For example, with the mixture ratio giving maximum power, the exhaust gas of an automobile engine may contain as much as 8 per cent of CO by volume, and corresponding amounts of other unburned combustible components. Modifications of the Orsat apparatus are therefore necessary if such an exhaust gas is to be analyzed accurately.

In the *modified Orsat* apparatus,[2] the CO_2 and O_2 are absorbed by the same reagents previously described; the proportions of CO and H_2 are determined by combustion to CO_2 and H_2O respectively; and measurement is made of the reduction in volume caused by the removal of these products of combustion. Saturated hydrocarbons, such as methane and ethane, are removed by slow combustion in contact with an electrically heated platinum wire, and the proportions are computed from measurements of the reduction in volume when the products of combustion are removed. Unsaturated hydrocarbons, or the illuminants, may be removed by contact with fuming sulphuric acid. (For more detailed instructions for making analyses of the exit gases see the references given.)

384. Properties of Gas Mixtures.—(a) Certain characteristics of gas mixtures need to be understood before proceeding with a diagnosis of the gas analysis; hence, there will now be given a brief review of the subject, which is more fully treated in Chap. VII, Part I.

(b) The **general equation** for a gas mixture is

$$P_m V_m = w_m R_m T_m, \quad . \quad . \quad . \quad . \quad . \quad (533)$$

in which the *gas constant* of the mixture is

$$R_m = \Sigma(wR) \div w_m.$$

TABLE XLIII

	Molecular Formula	Molecular Weight (Approximate)	R	$R \times$ Molecular Weight
Air	....	(29)	53.3	1547
Carbon monoxide	CO	28	55.3	1550
Hydrogen	H_2	2	762.0	1525
Nitrogen	N_2	28	55.3	1550
Oxygen	O_2	32	48.2	1544
Carbon dioxide	CO_2	44	35.0	1540
Acetylene	C_2H_2	26	59.9	1558
Methane	CH_4	16	96.5	1544
Ethylene	C_2H_4	28	55.3	1550
Steam at low pressure	H_2O	18	85.7	1544

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[2] Bureau of Mines Bull. No. 320, by Fieldner, Jones and Holbrook, and Research Paper No. 266, by Martin Shepherd: also Experimental Mech. Engrg. by Diederichs and Andrae, Chap. XII.

This equation may be used to find either the pressure, volume, weight, or temperature of a gaseous mixture when the other three variables and the weight constituents of the gaseous mixture are known. The values of R and the approximate molecular weights for the different individual constituent gases are given in Table XLIII.

If the molecular weight of a gas (or the equivalent molecular weight of a gas mixture) be known, a satisfactory value of R may be calculated by dividing 1544 by the molecular weight.

(**c**) The **specific heats** at constant pressure and constant volume, c_{p_m} and c_{v_m}, respectively, of the gaseous mixture may be found as follows:

$$c_{p_m} = \Sigma(wc_p) \div w_m \quad . \ . \ . \ . \ . \ . \ . \quad (534)$$

$$c_{v_m} = \Sigma(wc_v) \div w_m. \quad . \ . \ . \ . \ . \ . \ . \quad (535)$$

Instantaneous values of c_p and c_v for the constituent gases, at any given temperature, may be found from Plate VIII of the Appendix.

If the variation of specific heat with temperature is to be considered, then

$$c_{pm} = M'_m + B_mT_m + C_mT_m^2, \quad . \ . \ . \ . \quad (536)$$

and

$$c_{vm} = M_m + B_mT_m + C_mT_m^2, \quad . \ . \ . \ . \ . \ . \quad (537)$$

in which

$$M'_m = \Sigma[(wM') \div w_m]; \ M_m = \Sigma[(wM) \div w_m];$$

$$B_m = \Sigma[(wB) \div w_m]; \text{ and } C_m = \Sigma[(wC) \div w_m].$$

The values of M', M, B, and C for the constituent gases are given in Table E of the Appendix. Also, the mean values of the respective specific heats over a given temperature range may be estimated from Plate VIII of the Appendix.

(**d**) A mixture of individual gases having no chemical affinity obeys **Dalton's Law**; i.e., the total pressure, p_m, exerted by the mixture is the sum of the partial pressures of the constituent gases, and each constituent gas occupies the entire volume of the mixture.

(**e**) The **volumetric analysis** of the exit gases, such as is obtained by the Orsat apparatus, is based fundamentally on the foregoing principles. Thus, for example, if gases having the respective weights w_1, w_2, w_3, and w_4 lb. form a mixture of weight $w_m = w_1 + w_2 + w_3 + w_4$

lb. which occupies the volume V_m cu. ft. at the total pressure P_m lb. per sq. ft. and absolute temperature T_m deg. fahr., then

$$P_m = P_1 + P_2 + P_3 + P_4 = w_m R_m T_m \div V_m,$$

and

$$P_1 V_m = w_1 R_1 T_m;\ P_2 V_m = w_2 R_2 T_m;$$

$$P_3 V_m = w_3 R_3 T_m;\ P_4 V_m = w_4 R_4 T_m.$$

If each component gas, in turn, be exposed to the total mixture pressure, P_m, at the same temperature T_m (as in determining the constituents with the Orsat apparatus), then the respective gases will occupy volumes as follows:

$$V_1 = w_1 R_1 T_m \div P_m;\ V_2 = w_2 R_2 T_m \div P_m;$$

$$V_3 = w_3 R_3 T_m \div P_m;\ V_4 = w_4 R_4 T_m \div P_m;$$

and

$$V_m = V_1 + V_2 + V_3 + V_4.$$

Thus, the *volume ratio*, or the *fraction by volume*, of the first gas in the mixture is $f_1 = V_1/V_m = P_1/P_m$; and similarly for the other component gases: $f_2 = V_2/V_m = P_2/P_m$; $f_3 = V_3/V_m = P_3/P_m$; $f_4 = V_4/V_m = P_4/P_m$. It is these fractional quantities (f) that are given in reporting the volumetric analysis of the gas mixture as obtained with the Orsat apparatus.

(f) The **gravimetric** or **weight analysis** is often needed and may be found directly from the volumetric analysis. This analysis is based on the ratios of the weights of the constituent gases to the weight of the mixture. Thus, for example, the *gravimetric* or *weight ratio* for constituent No. 1 is

$$g_1 = \frac{w_1}{w_m} = \frac{P_1 R_m}{P_m R_1} = \frac{V_1 R_m}{V_m R_1} = f_1 \frac{R_m}{R_1}, \quad . \quad . \quad . \quad (538)$$

Also, if the equivalent molecular weight of the gas mixture be designated by m_m, where $m_m = \Sigma(fm)$, then $m_m R_m = m_1 R_1$, and the weight ratio becomes

$$g_1 = f_1 \frac{m_1}{m_m}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (539)$$

Similar equations can be written, of course, for the other constituents.

In words, the fraction *by weight* of each individual gas in a mixture is equal to the fraction of that gas by volume, multiplied either by the ratio of the gas constant of the mixture to the constant of the constituent, or by the ratio of the molecular weight of the constituent to the molecular weight of the mixture.

(g) As a **numerical application** of the foregoing equations consider a flue gas yielding the following volumetric analysis: $CO_2 = 0.125$, $CO = 0.005$, $O_2 = 0.07$, $N_2 = 0.80$; and assume the temperature of the mixture to be 500 deg. fahr. Then, taking the values of the molecular weights and of R from Table XLIII, the computed values for the gravimetric analysis and the specific heats c_p of the component gases at the given temperature, are as given in Table XLIV. (The values of c_p obtained by computation can be checked by means of Plate VIII in the Appendix). From the derived quantities tabulated, the following values are found for the mixture: $m_m = \Sigma(mf) = 30.28$, $R_m = \Sigma(gR) = 51.09$, and $c_{p_m} = \Sigma(gc_p) = 0.246$. The value of R_m may be roughly checked by dividing 1544 by the molecular weight of the mixture. The fifth column gives the gravimetric analysis.

TABLE XLIV

Gas	Volume Fraction (Given)	Molecular Weight	Weight		Gas Constant	gR	c_p at 500 deg. fahr.	gc_p
			Relative	Fraction				
	f	m	mf	$g=\frac{mf}{m_m}$	R			
CO_2	0.125	44	5.50	0.1814	35.0	6.35	0.235	0.0426
CO	0.005	28	0.14	0.0046	55.3	0.25	0.251	0.0012
O_2	0.070	32	2.24	0.0740	48.2	3.57	0.220	0.0163
N_2	0.800	28	22.40	0.7400	55.3	40.92	0.251	0.1860
Mixture	1.00 (Σf)		30.28 $\Sigma(mf)$	1.000 (Σg)		51.09 $\Sigma(gR)$		0.2461 $\Sigma(gc_p)$

385. Determination of the Weight of Exit Gas.—(a) The weight of *dry* exit gas formed from burning one pound of fuel may be determined from either a gravimetric or a volumetric analysis of the dry gas. The advantage of considering the gravimetric analysis is that it simplifies the explanation of this determination. On the other hand, the analysis of the exit gas is actually made on the volume basis, and it is simpler to use the developed equations if they are expressed in terms representing the fractional *volumes* of the constituent gases. Therefore, the explanations will all be based on the gravimetric analysis, but the derived equations will be rewritten on the volumetric basis. The methods of determining the weight of dry gas formed from the burning of one pound of fuel in a *furnace* and in an *internal combustion engine* will be treated separately. In each case, however, the respective weights of carbon in each pound of the exit gas and in each pound of the fuel are made the basis of the determinations.

A. WEIGHT OF DRY EXIT GAS FROM A FURNACE

(**b**) The carbon in the fuel burned in a furnace is found in the CO_2 and the CO of the exit gas; and, using the proper chemical symbols to represent the *fractional weights* of these constituent gases in the dry flue gas, the *weight of carbon in one pound of dry flue gas is*

$$C_g = (12/44)\ CO_2 + (12/28)\ CO.$$

Then, if C_b represents the weight of carbon actually burned per pound of *fuel fired*, it follows that, on *the gravimetric* basis, the **weight of the dry flue gas** formed *per pound of fuel fired* is

$$w_g = \frac{C_b}{C_g} = \frac{C_b}{(12/44)\ CO_2 + (12/28)\ CO} \text{ (by weights)}. \quad . \quad (540a)$$

Since the usual dry flue gas analysis shows only CO_2, CO, O_2, and N_2, then $CO_2 + CO + O_2 + N_2 = 1.0$, and Eq. (540*a*) might be written as follows:

$$w_g = \frac{C_b(CO_2 + CO + O_2 + N_2)}{(12/44)\ CO_2 + (12/28)\ CO} \text{ (by weights)}. \quad . \quad (540b)$$

(**c**) In order to put Eq. (540*b*) on the *volumetric basis*, the fractional weight, g, of each of the component gases may be replaced by the product of its fractional volume, f, and its molecular weight, m, divided by the molecular weight of the mixture, m_m; i.e., $g = fm \div m_m$. Then canceling the common term, m_m, and using the chemical symbols to represent the *fractional volumes* of the constituents, the **weight of dry flue gas** formed *per pound of fuel fired* becomes

$$w_g = \frac{C_b(44\ CO_2 + 28\ CO + 32\ O_2 + 28\ N_2)}{12\ CO_2 + 12\ CO},$$

$$= \frac{C_b(3.67\ CO_2 + 2.33\ CO + 2.67\ O_2 + 2.33\ N_2)}{CO_2 + CO} \text{ (by vol.)}, \quad . \quad (541)$$

in which the volume fractions are those found with the Orsat apparatus.

(**d**) As **an illustration** of these two methods of calculating the weight of dry flue gas per pound of fuel fired, consider the analysis of the flue gas to be that given in Table XLIV, and assume that the fuel contained 70 per cent carbon, by weight, and that tests show that 4.3 per cent of this carbon did not burn but was collected with the mineral ash. Then $C_b = 0.70\ (1 - 0.043) = 0.67$.

Using the *weight* fraction method, the weight of dry flue gas, w_g, is found from Eq. (540*a*) as follows:

$$w_g = \frac{0.67}{(3/11)0.1814 + (3/7)0.0046} = \frac{0.67}{0.0514} = 13.$$

Using the *volume* fraction method given in Eq. (541),

$$w_g = \frac{0.67(3.67 \times 0.125 + 2.33 \times 0.005 + 2.67 \times 0.07 + 2.33 \times 0.80)}{(0.125 + 0.005)}$$

$= 13$ lb. dry flue gas per pound of fuel fired.

B. WEIGHT OF DRY EXIT GAS FROM AN INTERNAL COMBUSTION ENGINE

(e) An analysis of the dry *exhaust gas* from an internal combustion engine will usually show the presence of N_2, CO_2, CO, H_2, O_2, and CH_4. In this case, using chemical symbols to represent the fractional weights of the constituents, the weight of carbon in one pound of the dry exhaust gas is

$$C_g = (12/44)\,CO_2 + (12/28)\,CO + (12/16)\,CH_4.$$

Then, if C_b is the weight of carbon burned per pound of fuel used, the **weight of dry exhaust gas** per pound of this fuel is, on the gravimetric basis,

$$w_g = \frac{C_b}{C_g} = \frac{C_b}{(3/11)\,CO_2 + (3/7)\,CO + (3/4)\,CH_4} \text{ (by weights)}. \quad (542a)$$

Also, since $N_2 + CO_2 + CO + H_2 + O_2 + CH_4 = 1.0$,

$$w_g = \frac{C_b(N_2 + CO_2 + CO + H_2 + O_2 + CH_4)}{(3/11)\,CO_2 + (3/7)\,CO + (3/4)\,CH_4} \text{ (by weights)}. \quad (542b)$$

In order to put this equation for the weight of dry exhaust gas on the *volumetric basis* it is necessary to substitute for each *fractional weight*, g, its equivalent, $fm \div m_m$, as before. Using the chemical symbols to represent fractional volumes, Eq. (542*b*) becomes

$$w_g = \frac{C_b(28\,N_2 + 44\,CO_2 + 28\,CO + 2\,H_2 + 32\,O_2 + 16CH_4)}{12(CO_2 + CO + CH_4)}. \quad (543)$$

If the weight of carbon burned per pound of fuel used (C_b) is not directly available for use in these equations, it can be closely approximated by calculations based on the exhaust gas analysis.

(f) **Based on the carbon content of the exhaust gas,** the determination of the **weight of carbon burned** and of the **weight of dry exhaust gas formed** per pound of fuel used is as follows:

The total weight of combustible of the fuel accounted for in one pound of the exhaust gas is the sum of the weights of carbon and hydrogen represented. The hydrogen in the exhaust gas is partly in the water vapor and partly unburned. The weight of hydrogen burned is one-eighth of the weight of oxygen combined with hydrogen; but, the weight of oxygen combined with hydrogen is the difference between the weight of oxygen supplied and the weight of the oxygen accounted for in the *dry gas* analysis, since the water vapor formed by the reaction of H_2 and O_2 does not appear in such an analysis. Repeating these statements in a mathematical form, still using fractional weights, the **weight of combustible** in the fuel, as represented by *one pound of the* ***exhaust gas,*** is, on the *gravimetric basis,*

$$B_g = (12/44)\ CO_2 + (12/28)\ CO + H_2 + CH_4 + 1/8\ [N_2/3.31 - (32/44)\ CO_2 - (16/28)\ CO - O_2]) \quad . \quad . \quad (544)$$

Thus, the **fraction of carbon,** by *weight,* in the *combustible* of the fuel is then,

$$C_b = C_g/B_g$$

$$= \frac{(12/44)\ CO_2 + (12/28)\ CO + (12/16)\ CH_4}{(8/44)\ CO_2 + (10/28)\ CO + H_2 + CH_4 + N_2/26.48 - O_2/8} \quad . \quad (545)$$

Then, *assuming the fuel to be entirely combustible,* the **weight of dry exhaust gas** ***per pound of fuel*** used is

$$w_g = C_b/C_g$$

$$= \frac{1}{(8/44)\ CO_2 + (10/28)\ CO + H_2 + CH_4 + N/26.48 - O_2/8} \quad . \quad (546)$$

By introducing fractional volumes into Eq. (545) and assuming the fuel to be all combustible, the **weight of carbon** burned per pound of the *fuel* used is, on the *volumetric basis,*

$$C_b = \frac{12(CO_2 + CO + CH_4)}{8\ CO_2 + 10\ CO + 2\ H_2 + 16\ CH_4 + 1.057\ N_2 - 4\ O_2} \quad . \quad . \quad (547)$$

By similarly changing Eq. (546), the expression for the **weight of dry exhaust gas** then becomes, on the *volumetric basis*,

$$w_g = \frac{m_m}{8\,CO_2 + 10\,CO + 2\,H_2 + 16\,CH_4 + 1.057\,N_2 - 4\,O_2}$$

$$= \frac{28\,N_2 + 44\,CO_2 + 28\,CO + 2\,H_2 + 32\,O_2 + 16\,CH_4}{8\,CO_2 + 10\,CO + 2\,H_2 + 16\,CH_4 + 1.057\,N_2 - 4\,O_2} \quad . \quad . \quad (548)$$

(g) As an **illustration** of the use of the volumetric equations in order to find the weight of dry exhaust gas formed from the combustion of one pound of fuel in an internal combustion engine, assume the following typical *volumetric analysis* of the dry exhaust gas:

$CO_2 = 0.0896$ $\quad H_2 = 0.0300$
$O_2 = 0.0146$ $\quad CH_4 = 0.0100$
$CO = 0.0700$ $\quad N_2 = 0.7858$

The weight of carbon burned per pound of the fuel used in this engine is, from Eq. (547),

$$C_b = \frac{12 \times 0.1696}{2.4094} = 0.845 \text{ lb. per lb. of fuel.}$$

The weight of dry exhaust gas per pound of fuel used is, from either Eq. (546) or Eq. (548), 11.86 lb.

C. WEIGHT OF WET EXIT GASES

(**h**) The total weight of exit gas *including the water vapor content* may be found by adding to the weight of dry exit gas per pound of fuel, w_g, the weight of water vapor in the gas. This water vapor comes from: (1) the total moisture carried by the fuel,[3] and in amount equal to $1\frac{1}{8}w_O$; (2) the water vapor carried by the air supplied for combustion, as given by a psychrometric analysis of the air; (3) the water vapor formed from the hydrogen actually burned. The weight of *water vapor coming from the last source* per pound of fuel is

$$w_{vh} = 9[w_{AH} - (2\,H_2 \div m_m)w_g], \quad . \quad . \quad . \quad . \quad (549)$$

in which w_{AH} = weight of available hydrogen per pound of fuel,[3]
$= w_H - w_O/8$.
H_2 = the fraction, by volume, of unburned hydrogen in the dry exit gas.
m_m = the molecular weight of the exit gas.
w_g = the weight of dry exhaust gas per pound of fuel.

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[3] The *total moisture carried by the fuel* is the sum of the free or hygroscopic moisture, given in the proximate analysis, and the water of combination. It is commonly assumed that all of the oxygen, w_O, reported in the ultimate analysis, is combined with a portion of the hydrogen in the form of water of combination. The weight of this water of combination, per pound of fuel, is $1\frac{1}{8}\,w_O$. Then the *available hydrogen* in one pound of fuel, w_{AH}, is the difference between the total hydrogen and the hydrogen combined with oxygen, or $w_{AH} = w_H - w_O/8$.

386. Finding the Weight of Air Supplied.—(**a**) The *amount of air* supplied for combustion can be determined either by metering it before use, or by deduction from the analysis of the exit gases. The first method involves considerable expense and difficulty, hence the second is the one more commonly followed, especially when large amounts of air are to be measured.

(**b**) When **coal** is the fuel burned, the assumption may be made that all of the N_2 in the exit gas is supplied from the air; then, since nitrogen constitutes 76.8 per cent of the air by weight, the *weight of air supplied* for combustion per pound of this fuel burned is

$$w_A = \frac{28\ N_2 \div 0.768}{12(CO_2 + CO)} C_b = \frac{3.04\ N_2}{CO_2 + CO} C_b \quad \text{(by vol.)}. \quad . \quad . \quad (550)$$

For example, in the case of the flue gas and coal previously considered in Sects. **384** (**g**) and **385** (**d**).

$$w_A = \frac{3.04 \times 0.8}{0.13} \times 0.67 = 12.5 \text{ lb. of air per lb. of } \textit{coal}.$$

(**c**) For a **liquid fuel** containing little or no nitrogen, such as that generally used in internal combustion engines, the weight of air supplied is equal to the weight of air represented by the N_2 in the exhaust gas divided by the weight of fuel used, as represented in the exhaust gas by B_g in Eq. (544). Using fractional *weights* in the equation, this *weight of air per pound of fuel* used, becomes

$$w_A = \frac{N_2/0.768}{(8/44)\,CO_2 + (10/28)\,CO + CH_4 + H_2 + N_2/26.48 - O_2/8}. \quad (551)$$

Using fractional *volumes* of the constituent gases, the weight of air supplied per pound of combustible, or the mixture ratio, is

$$w_A = \frac{36.5\ N_2}{8\ CO_2 + 10\ CO + 16\ CH_4 + 2H_2 + 1.057\ N_2 - 4\ O_2}. \quad . \quad (552)$$

Thus, for example, with the automotive engine exhaust gas analysis previously given in Sect. **385** (**g**), the mixture ratio is

$$w_A = \frac{36.5 \times 0.7858}{2.4094} = 11.9 \text{ lb. air per lb. of fuel.}$$

(**d**) *If the weight of ash (if any) and the weight of exit gas per pound of fuel be known, the weight of air supplied can then be ascertained directly, without further recourse to the exit gas analysis.*

Thus, the total weight of the wet exit gas per pound of fuel is

wet gas = dry gas + water vapor = air + fuel − ash;

or, in symbols,

$$w_w = w_g + w_v = w_A + w_{\text{fuel}} - w_{\text{ash}}.$$

Thus, the weight of air supplied per pound of fuel is

$$w_A = w_g + w_v + w_{\text{ash}} - 1 \qquad (553a)$$

$$= w_w + w_{\text{ash}} - 1. \qquad (553b)$$

If the fuel is *liquid* or *gaseous*, without inert solid matter, this equation still applies, the ash being zero.

(e) The **weight of air just necessary** for the complete combustion of *any kind* of fuel may be calculated by the method explained in Sect. **373** if the ultimate analysis of the fuel is known. When the weight required *per pound* of any kind of fuel is desired, its value is given by w_a in Eq. (524).

If the fuel is *gaseous*, and its composition is expressed in terms of the fractional weights of component combustibles in one pound of the mixture, then, letting the symbols represent these *fractional weights*, the weight of air necessary for complete combustion is

$$w_a = 2.46w_{\text{CO}} + 34.5\,(w_{\text{H}} - w_{\text{O}}/8) + 17.2\,w_{\text{CH}_4} + 13.3\,w_{\text{C}_2\text{H}_2} + 14.8\,w_{\text{C}_2\text{H}_4} + 16.1w_{\text{C}_2\text{H}_6}. \qquad (554)$$

The relation between w_A and w_a has an important bearing on the actual combustion of fuels and will be discussed in the next two sections.

387. Excess Air and CO_2 Content of Exit Gas.—(a) If a flue gas analysis shows no CO or other combustible matter present, it indicates at once that the reactions are complete; if at the same time free O_2 is present, it follows that the air supplied for combustion exceeded the amount just necessary for complete combustion. If the analysis shows that the exit gas contains combustible matter but no free O_2, the air supply was deficient. If both combustible matter and oxygen are present, it may be concluded that (1) sufficient air was *probably* supplied, but either (2) that the fuel and air were not properly mixed at all points in the combustion zone, (3) that combustion was imperfect because of low temperature at some points in that region, (4) that there was not sufficient time, or length of flame travel, allowed for the gases to burn out, (5) that air infiltration occurred, as through leaky furnace and boiler walls or flue joints, or (6) that a combination of these factors existed. Such general information is an important aid in the regulation of the combustion process and in maintaining boiler settings and flues in proper condition.

In order to avoid the possibility of incomplete reactions occurring at any point in the combustion zone because of unequal distribution (stratification) of the oxygen due to lack of intimate mixing, it is necessary to supply a furnace with a quantity of air somewhat in excess of the amount just necessary with perfect distribution and mixing. This results in an excess weight of flue gas and is a source of some heat waste, as will be seen later, but this loss is only a small fraction of that which might result from having insufficient air. In internal combustion engines, the mixture ratio which gives *maximum economy* at high fractional engine loads, is considerably leaner than the mixture ratio required for complete combustion. The relation between the amount of air actually supplied and that just necessary for complete combustion with perfect mixing is given by the **dilution ratio,** the **dilution coefficient,** or the **excess coefficient,** D, where

$$D = \frac{\text{weight of air actually used per lb. of fuel}}{\text{weight just needed for complete combustion of 1 lb. of fuel}} = \frac{w_A}{w_a} \quad . \quad (555)$$

(**b**) With **pure carbon** as the fuel undergoing complete combustion, the relation between the dilution coefficient, D, and the CO_2 content may be easily derived. From the coefficients in the chemical reaction equation for carbon burning to CO_2, it is seen that each reacting volume of oxygen forms an equal volume of CO_2, when measured at the same temperature and pressure. Since the air is 20.9 per cent O_2 by volume, the products of combustion will consist of 20.9 per cent CO_2, *by volume, when there is no excess and no deficiency of air.* In this one case, the dilution coefficient is unity. With excess air, however, the fraction, *by volume,* of CO_2 in the flue gas becomes

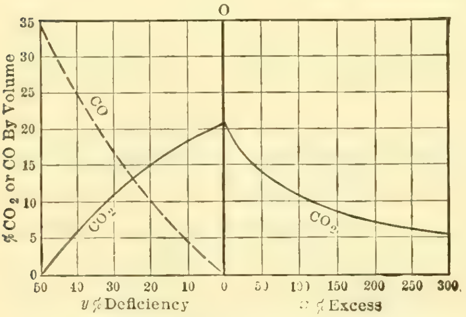

FIG. 533.—CO_2 and CO Content of Products of Combustion of Pure Carbon.

$$CO_2 = 0.209 \div D. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (556)$$

The **percentage of excess air,** x, is readily found from the dilution coefficient; *in general,* the relation is

$$x = 100\,(D - 1). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (557)$$

When *pure carbon* is the fuel, the per cent excess air is

$$x = 100(0.209/CO_2 - 1) \text{ (by vol.)}. \quad . \quad (558)$$

How the fraction, by volume, of CO_2 in the products of combustion of *pure carbon* varies with x and D, in accordance with Eq. (558), is shown numerically in Table XLV and graphically in Fig. 533. The figure also shows the percentage volumes of CO_2 and CO as a function of the per cent deficiency of air, y.

TABLE XLV

COMBUSTION OF PURE CARBON

D	1.0	1.5	2.0	3.0	4.0
x	0	50	100	200	300
Per cent CO_2, by volume of	20.9	13.9	10.45	6.97	5.23

(c) If **any actual fuel,** which contains other combustibles than carbon, be burned with a given quantity of excess air, the fractional volume of CO_2 in the products of combustion is *not* given by Eq. (558). Nor can this equation be solved to find the per cent excess air and hence the dilution ratio from the CO_2 content, if the fuel is other than pure carbon.

However, if the air, w_A, supplied for the combustion of one pound of the *actual* fuel, has been determined by measurement, or by calculations from the exit gas analysis (as in Sect. **386**), and if the fuel analysis is known, the *dilution coefficient* becomes, from Eqs. (524) and (555),

$$D = w_A \div [11.5w_c + 34.5(w_H - w_O/8) + 4.32w_S], \quad . \quad . \quad (559)$$

in which the symbols represent fractions, by *weight*, of the elements in one pound of the fuel. Having D, the per cent excess air may then be found from Eq. (557). If the composition of the fuel is unknown, the excess air and dilution coefficient are best determined from the oxygen content of the exit gas. This determination is explained in the next section.

How the percentage by volume of CO_2 in the products of combustion varies with the amount of excess air and also with different actual fuels is shown [4] in Fig. 534. Evidently it is not correct to use the per cent, by volume, of CO_2 in the exit gas as a basis for comparing the amounts of excess air supplied for combustion, unless the *same fuel* is used in each case involved. For example, with constant excess air of 50 per cent, the per cent by volume of CO_2 in the products of combustion of coke is shown to be 13.9, of natural gas, 7.3 per cent, and of blast furnace gas, 20.3 per cent.

(d) For any **one fuel,** however, **the CO_2 content of the exit gas may be used as an index of the effectiveness** of combustion. But, high CO_2 content does *not* necessarily mean complete combustion, because there may be at the same time a large quantity of unburned combustible, such as CO, in the exit gas. The CO_2 content which gives the best results, all things considered, is largely dependent upon the chemical constitution of the fuel, the manner of the burning, and the method of utilizing the heat developed.

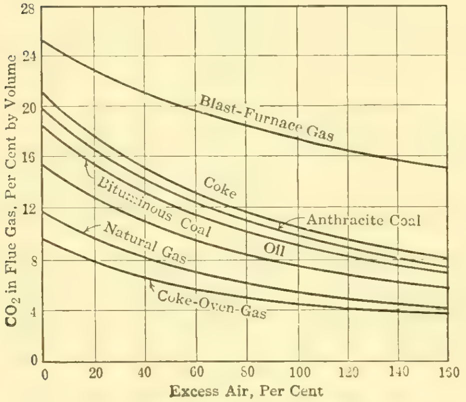

FIG. 534.—CO_2 Content of Products of Combustion of Various Specific Fuels.

In general, for any one fuel, the higher the CO_2 content of the exit gas becomes, the less will be (1) the excess air, (2) the weight of the exit gas per pound of fuel burned, and (3) the exit gas temperature; consequently, the smaller will be the loss of energy in the dry flue gases and in the superheated water vapor carried by these. On the other hand, the imperfections in the mixing, ignition, and time of contact between the fuel and oxygen may become so serious when the excess air decreases (i.e., CO_2 increases) unduly, that there may be an increased amount of unburned combustible in the exit gas, occasioning an *increase* in the loss due to incomplete combustion. There are, then, **optimum values of the CO_2 content and the dilution coefficient** with each fuel and set of operating conditions, above or below which it is undesirable to go. This is well illustrated in Fig. 535, which is for the combustion of natural gas. [5] In this specific case the lowest total combustion loss is seen to occur when the CO_2 content is about 10 per cent and the dilution coefficient is about 1.15; these values would therefore be the optimum ones in this case. With other fuels or other conditions similar curves may be drawn, but their positions and the optimum values will be different.

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[4] From "Limiting Factors in Reducing Excess Air in Boiler Furnaces," by E. G. Bailey; Mech. Engrg., July, 1926, p. 703.

[5] Fig. 535 is obtained from data in Proc. N.E.L.A., 1929, p. 1171.

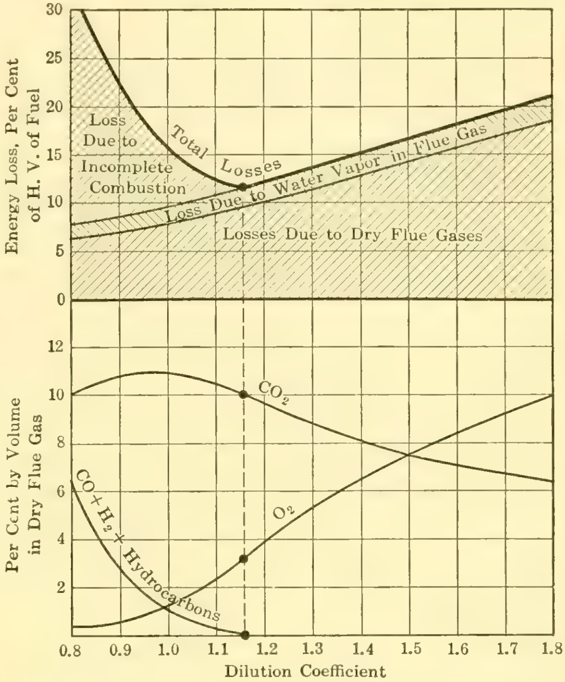

FIG. 535.—The Effect of Dilution Coefficient and CO_2 upon the Losses in a Steam-Generating Unit using Natural Gas.

388. Excess Air and O_2 Content of Exit Gases.—(a) In the preceding section it was seen that the CO_2 contents of the exit gases, by themselves, are not always satisfactory indices of the excess air supplied for combustion, particularly so when comparing combustion with different fuels. On the other hand, the fractional volume of O_2 (or of the N_2 which accompanies it in atmospheric air) is a fairly accurate criterion of excess air in the case of different fuels, because this O_2 content remains more nearly constant for different fuels when burned with a given percentage of excess air. Provided the fuel contains no appreciable amount of nitrogen, the dilution coefficient or the excess air may be computed from the exit gas analysis, largely on the basis of the oxygen content of this gas.

(b) For **flue gases from furnaces,** in which the only combustible constituent commonly present in appreciable amounts is CO, the determination of D and x is as follows: Using chemical symbols to represent the *fractional volumes* of the constituent gases, a volume of oxygen equal to 0.5 CO would be required to complete the combustion. Then if O_2 is the amount of free oxygen in the flue gas, the true excess oxygen is (O_2 − 0.5 CO), and the volume of nitrogen associated with this amount of oxygen in air is 3.78 times as much. The volume of nitrogen accompanying the oxygen chemically required for complete combustion is $N_2 - 3.78\ (O_2 - 0.5\ CO)$. Assuming that all of the nitrogen in the flue gas comes from the air (none from the fuel), the *dilution coefficient* may then be found from

$$D = \frac{N_2}{N_2 - 3.78(O_2 - 0.5\ CO)}\ \text{(by vol.)}. \quad . \quad . \quad . \quad . \qquad (560)$$

Similarly, the *excess air percentage* is found from

$$x = 100\,\frac{O_2 - 0.5\ CO}{0.264\ N_2 - (O_2 - 0.5\ CO)}\ \text{(by vol.)}. \quad . \quad . \qquad (561)$$

An alignment chart, as prepared by Professor C. O. Mackey, for the solution of this equation is shown in Fig. 536 and this gives a satisfactory approximate value of the excess air directly from the volumetric analysis of the flue gas, when the fuel itself does not contain appreciable quantities of nitrogen, and when the exit gas contains no combustible other than carbon monoxide.

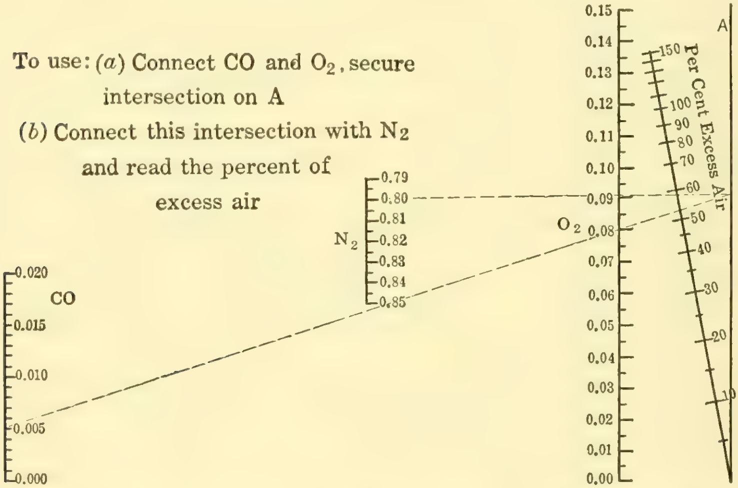

FIG. 536.—A Chart for Determining Excess Air from Volumetric Analysis of Flue Gas. (See Eq. 561.)

(c) The **exhaust gas from an internal combustion engine** may contain H_2 and CH_4, in addition to CO, as combustibles. In this case, if free O_2 is also present, the true excess oxygen is $(O_2 - 0.5\ CO - 0.5\ H_2 - 2CH_4)$. If there is no nitrogen in the fuel, the dilution coefficient is closely given by

$$D = \frac{N_2}{N_2 - 3.78(O_2 - 0.5\ CO - 0.5\ H_2 - 2\ CH_4)}\ \text{(by vol.)}. \quad . \qquad (562)$$

(**d**) As an illustration of the use of this equation, the dilution coefficient may be found directly for the automotive engine exhaust gas analysis given in Sect. **385** (**g**). In this case

$$D = \frac{0.7858}{0.7858 - 3.78(0.0146 - 0.035 - 0.015 - 0.02)} = 0.79.$$

To check this dilution coefficient using the weight of air supplied for combustion and the fuel analysis: w_A has been found in Sect. **386** (*c*) to be 11.9 lb. for this analysis. The fuel has been found, in Sect. **385** (*g*), to be 84.5 per cent carbon by weight, and must contain 15.5 per cent hydrogen by weight. The air just necessary for complete combustion of one lb. of this fuel is

$$w_a = 11.5 \times 0.845 + 34.5 \times 0.155 = 15.1 \text{ lb.}$$

By this method, the dilution coefficient is

$$D = w_A/w_a = 11.9/15.1 = 0.79, \text{ as before.}$$

389. The Exit Gas Analysis As An Indication of Losses.—(a) In analyzing the performance of a furnace or an internal combustion engine, the exit gas analysis may be used to determine the nature and magnitude of three important losses of energy. These losses are:

(1) The loss due to the energy carried away by the dry exit gases leaving at a temperature above that of the air supplied for combustion.

(2) The loss due to incomplete burning of the combustible elements of the fuel. This includes the loss due to the presence of unburned carbon (as in smoke) and of CO, H_2 and hydrocarbons.

(3) The loss due to the energy carried away by the water vapor leaving with the exit gases at a temperature above that of the air and fuel.

(**b**) The magnitude of the first loss, **the dry gas loss**, depends upon the weight of the dry exit gas formed per pound of fuel burned, the temperature elevation, and the specific heat of the dry products of combustion. In mathematical form, the quantity of heat lost in the dry gas, in B.t.u. per pound of fuel burned, is

$$Q_d = \bar{c}_p w_g(t_x - t_a,) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (563)$$

in which $\bar{c}_p$ is the mean specific heat at constant pressure of the gas mixture, t_x is the exit gas temperature, t_a is the temperature of the air supplied for combustion, and w_g is the weight of dry exit gas per pound of fuel burned. The specific heat of the exit gas may be determined by the method given in Sect. **384**; more commonly, however, a value of 0.24 B.t.u. per lb. per deg. fahr. is used. The weight of dry gas is determined by the method given in Sect. **385**.

(c) The magnitude of the **loss due to incomplete combustion** depends upon the weight and heating value of the combustible matter present in the exit gas. This loss, in B.t.u. per pound of fuel burned, would be

$$Q_i = 14{,}150\ \mathrm{C} + 3960\ \mathrm{CO} + 61{,}000\ \mathrm{H_2} + 23{,}670\ \mathrm{CH_4}, \quad . \quad . \quad (564)$$

in which the chemical symbols represent the *weights* of the respective combustibles in the exit gas per pound of fuel burned.

The unburned carbon, appearing in Eq. (564), is not obtained directly from the exit gas analysis, hence this component must be determined from other data. Accurately, this unburned carbon, C, is the difference between the total carbon in one pound of fuel (as given in the ultimate analysis) and the sum of the carbon represented in the carbon compounds of the exit gas and the unconsumed carbon in the solid refuse (ashes) per pound of fuel burned. Frequently, however, because of the difficulty in determining accurately how much of the refuse from a coal fire is unburned carbon, and because the percentage involved is relatively small, the loss due to the presence of uncombined carbon in the exit gas is neglected when the combustion of coal is being studied.

(d) The **loss due to water vapor** is directly dependent, of course, on the amount of this vapor leaving with the exit gases. This vapor may come from one or more of the three following sources: (1) The moisture which is contained in the fuel, as fired, and which is vaporized in the furnace; (2) the water vapor formed from the combustion of hydrogen and hydrocarbons in the fuel; (3) the water vapor in the air supplied for combustion, the amount depending upon the relative humidity and temperature of the air.

All this vapor becomes highly heated in passing through the furnace or engine and leaves the apparatus as superheated steam, if the temperature of the exit gas is above the dew point of the gaseous mixture, as is usually the case. The heat required for all this vaporization and superheating occasions a very appreciable loss when the exit gases leave the energy-absorbing apparatus at a much higher temperature than that of the entering air and fuel.[6] However, even if the temperature of the exit gases were reduced to that of the entering air by contact with sufficient heat-absorbing surface or by expansion in an engine, all of this heat classed as a loss could not be recovered *unless* the entering air had a relative humidity of 100 per cent (i.e., was saturated with water vapor). If the air supplied for combustion were unsaturated, a portion, at least, of the water vapor formed from the moisture and hydrogen in the fuel would remain as superheated steam at the exit gas temperature, saturating the products of combustion. In this case, a portion of the latent heat of vaporization would still be lost, even if the products of combustion were returned to the temperature of the entering air, as is done in the determination of the calorimeter heating value of a fuel.

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[6] Although the H_2O in the exit gas from steam-generating equipment or from internal combustion engines is a source of heat loss, and from this standpoint, undesirable, it may be utilized to advantage in the case of airships. Enough of the water vapor in the exhaust from the internal combustion engines driving such lighter-than-air ships can be condensed, in air-cooled condensers, to equalize the weight of the fuel burned, thereby avoiding the expensive practice of wasting hydrogen or helium by valving it from the gas bags in order to keep the ship at the proper level. See "Water Recovery Apparatus for Airships," by C. P. Burgess, Mech. Eng., July, 1931.

It has been, nevertheless, the common practice in this country to charge the steam-generating apparatus with the higher heating value of the fuel, and the gross loss is therefore calculated.

(1) The **loss due to the moisture in the fuel,** as fired, is equal to total weight of moisture in one pound of fuel, times the increase in the heat content of this moisture in passing through the apparatus involved, or

$$Q_m = w_m(h_x - h_f), \quad . \quad . \quad . \quad . \quad . \quad (565)$$

in which Q_m = loss of heat due to moisture in fuel, in B.t.u. per pound of fuel, as fired.

w_m = total weight of moisture [7] in one pound of fuel, as fired.

h_x = the heat content of the steam in the exit gas at the exit temperature t_x, in B.t.u. per pound.[8]

h_f = the heat content of the water at the temperature of the entering fuel, in B.t.u. per pound.

The heat content of superheated steam at pressures under 2 lb. per sq. in. abs. and at temperatures (t_x) between 200 and 600 deg. fahr. may be closely obtained by the empirical equation

$$h_x = 1057 + 0.46t_x.$$

At low temperatures, $h_f = t_f - 32$ (approximately).
Therefore, with these values, Eq. (565) becomes

$$Q_m = w_m(1089 + 0.46t_x - t_f). \quad . \quad . \quad . \quad . \quad (566)$$

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[7] See Sect. **385** (**h**).

[8] The place at which t_x is measured depends entirely upon the apparatus involved; but in any case, it should be at that point beyond which no further useful energy (either mechanical or thermal) is obtained from the products of combustion within the given apparatus, whether an internal combustion engine or a steam-generating unit.

(2) The **loss due to water vapor formed from available hydrogen in the fuel** (since there are 9 lb. of water vapor formed from each pound of hydrogen) is, in B.t.u. per pound of fuel, as fired,

$$Q_h = 9w_{AH}(h_x - h_f) \quad . \; . \; . \; . \; . \; . \; . \; . \quad (567a)$$

$$= 9w_{AH}(1089 + 0.46t_x - t_f), \quad . \; . \; . \quad (567b)$$

in which $w_{AH} = w_H - w_O/8$ = the weight of available hydrogen in one pound of fuel, as fired.

(3) The **loss due to the water vapor in the air supplied** is usually small and is sometimes neglected in the heat balance. The calculation of this loss involves the temperature and relative humidity of the air supplied for combustion. Both of these quantities may be found by the aid of a sling psychrometer, and the weight of water vapor carried by each pound of the dry air may be determined in the manner explained in Sect. **198**. The superheated water vapor in the atmosphere is at a very low pressure (usually under 0.5 lb. per sq. in. abs.), and its specific heat at constant pressure is substantially constant and equal to 0.46.

Consequently, *the loss due to the water vapor in the combustion air*, in B.t.u. per pound of fuel, as fired, becomes

$$Q_v = 0.46(t_x - t_a)w_v w_A, \quad . \; . \; . \; . \; . \; . \; . \quad (568)$$

in which w_v = the weight of vapor in one pound of dry air supplied for combustion, in pounds.
w_A = the weight of dry air supplied per pound of fuel burned, in pounds.
t_a = the temperature of the entering air, in deg. fahr.

(**e**) In reporting the results of a test of a boiler and furnace or of an internal combustion engine, **a heat or energy balance** is usually compiled. In the boiler furnace heat balance the higher heating value of one pound of the fuel is commonly accounted for by the combustion losses, the heat dissipated to the surroundings by heat transfer, the grate losses, and the heat supplied to the water and steam. In the heat balance for the internal combustion engine, the heat supplied is accounted for by the combustion losses, the heat given up to the jacket water, the heat lost to the surroundings by heat transfer, as from the bearings, and the energy delivered by the engine. In either case, the combustion losses may account for from 10 to 50 per cent of the heating value of the fuel.

390. The Dew Point and the Corrosive Action of Flue Gases.—(a) Some constituents of flue gas are present in such small amounts that they are not worth determining in the usual gas analysis, and even if their concentrations were known, the previous calculations would not be affected appreciably. However, these same constituents may cause serious corrosion of flues, air preheaters, economizer tubes, and other parts in the path of the exit gas. Such constituents are sulphur, SO_2, SO_3, and other impurities in the fuel and the products of their chemical reactions.

(**b**) Methods for determining the amounts of SO_2 and SO_3 present in flue gas have been developed, and the causes of corrosion in power plants have been carefully studied.[9] The results of the study indicate that the most serious corrosion of preheater and economizer tubes occurs when conditions are favorable to the formation of a liquid film on the surface of the metal exposed to the exit gases. If water vapor is the only vapor present in the exit gases, the temperature at which the exit gas is saturated, or *dew point*, can be determined by using the flue gas analysis.

(**c**) For example: Considering the volume analysis of the flue gas formed in the combustion of coal given in Sect. **384** (g), there is 0.8 fractional volume of N_2 appearing in the dry gas. Assuming that all of this nitrogen came from the air, there must have been $0.8 \div 3.78$ or 0.2115 fractional volume of O_2 accompanying this nitrogen. However, in the dry exit gas, only $(0.125 + 0.07 + 0.005 \div 2)$ or 0.1975 fractional volume of O_2 is accounted for. The difference, $(0.2115 - 0.1975)$, or 0.014 fractional volume of O_2 must have formed H_2O, which does not appear in the dry gas. This amount of O_2 would form 2×0.014 or 0.028 fractional volume of H_2O. The fraction, by volume, of water vapor in the wet products of combustion, at or above the dew point, is $0.028 \div 1.028$ or 0.027, neglecting the water vapor supplied in the air. At or above the dew point, 0.027 is also the ratio of the partial pressure of the water vapor to the total pressure of the wet exit gas, again neglecting the water vapor supplied in the combustion air. Then, if the total pressure of the wet exit gas is, say, 14.5 lb. per sq. in. abs., the partial pressure of the water vapor is 0.027×14.5 or 0.39 lb. per sq. in. abs. The saturation temperature at this pressure is about 72 deg. fahr. This temperature is therefore the dew point of the exit gases, for condensation of water would occur if the gases were cooled to this temperature. The *effect of more water vapor*, whether supplied in the combustion air, or by the burning of a fuel having a higher hydrogen content than the coal for which the exit gas analysis is given in this example, is to raise the dew point. Commonly, the partial pressure of the water vapor in flue gas will be such that the dew point will be around 100 deg. fahr. with coal as fuel.

(**d**) *If two or more condensable vapors* are present in a gas mixture, the dew point may then be defined as the highest temperature at which any of the corresponding liquids may exist in equilibrium with the gas. The effect of the presence of SO_3 in the flue gas, even in very small amounts, is to increase the dew point to as high as 200 deg. fahr. Thus if the temperature of the tube surface in an economizer or air preheater is near this value, a liquid film of sulphuric acid may be formed. This acid film rapidly corrodes iron, steel and other metals, but not lead.

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[9] See Bulletin No. 228, Univ. of Illinois, Engrg. Expt. Sta., "The Corrosion of Power Plant Equipment by Flue Gases," by H. F. Johnstone, June 9, 1931.

391. Atmospheric Pollution by Exit Gases.—(a) The atmosphere may be vitiated by both visible and invisible discharges from furnaces and internal combustion engines. The discharge is made visible by the presence of dark or opaque particles of solid matter which obscure the light, pollute the atmosphere and eventually become deposited, often causing annoyance or damage. Invisible discharges, although composed mainly of gases and vapors, may also contain solid matter which is of such fineness and color that it is not ordinarily observed. Some of the invisible material may be noxious (such as CO and SO_2) and some may be corrosive.

The discharge may (1) indicate a fuel loss due to incomplete combustion, (2) damage property, (3) affect health (due to its noxious contents, or due to its absorption of the ultra-violet rays from the sun), or cause depressing psychological conditions, and (4) retard or destroy vegetation. Closely associated with these effects are the economic losses involved; for example, the preventable money loss per capita in a smoky city has been variously estimated to be from $10 to $20 per year, due to expenditures occasioned for extra laundry work, cleaning, redecorating, reduction in life of building materials, loss of fuel, etc.[10]

(b) **Smoke** which issues *from furnaces* is composed chiefly of very small particles of carbon and tar carried by the gases and is ascribed to the decomposition of hydrocarbons at high temperature in the absence of oxygen. The actual amount of carbon in dense smoke is seldom more than one per cent of the carbon in the fuel, but its presence may indicate a considerable loss due to unburned *gaseous* combustible matter that is invisible. On the other hand, the absence of smoke does not necessarily indicate complete combustion, for combustion without smoke may, under unfavorable conditions, be even less complete than that with it. Smoke from furnaces can be avoided or minimized by using proper combustion equipment and methods of operation. Smoke from *internal combustion engines* may be due to using too rich a ratio of fuel to air, in which case the smoke is black, or to the discharge of lubricating oil vapor which is a blue-gray color.

Coarse soot or other solid matter, such as fly ash, emitted with the exit gases fall near their source; smaller particles are deposited at a distance depending on their size and on the velocity of the air currents; and very fine particles that are difficult to discern may float and form a smoke blanket in a stagnant atmosphere.

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[10] "Damage Due to Smoke," by Mellar, Trans. A.S.M.E., 1928, FSP 50–75, p. 213.

(c) Of the invisible exit gases, carbon monoxide is the most dangerous. If it is present, even in relatively small amounts, in the air breathed it may cause illness and possibly sudden death. The exhaust from an internal combustion engine may contain from 8 to 10 per cent CO by volume when the engine is operating on a rich mixture, and many deaths have resulted from running automotive engines in closed garages. The carbon monoxide in the exit gases may be eliminated or reduced to harmless amounts by proper regulation of combustion conditions, or in special cases, chemical absorption of this gas can be employed.

Sulphur products, when exhausted to the atmosphere, may have a detrimental influence on health as well as a harmful effect on vegetation and structural materials. As an indication of how serious the sulphuric problem may become, assume a community to burn annually 5 million tons of coal containing 3 per cent of sulphur; then the total sulphur in this coal is 150,000 tons, and the sulphuric acid equivalent is 459,000 tons per annum. Fortunately, however, most of the sulphur remains in the ash, and of that in the atmosphere only a very small fraction reaches ground level in the form of sulphuric acid.

(d) The **removal of solid material** from the flue gases is accomplished by such means as (1) mechanical separators (including screens, baffle traps, cyclone collectors, and rotating centrifugal separators), (2) electrical precipitators, and (3) hydraulic washers.[11] The **removal of sulphur products** from the flue gas is usually accomplished by washing the gases with water, as is done in the Battersea power station in London. Because of strong public opposition to sulphur pollution from this plant, extensive experiments [12] were conducted on methods of eliminating the SO_2 and SO_3 from the products of combustion, and as a result there was installed in the plant extensive washing apparatus using sprays of hot water (150 to 190 deg. fahr.), with provision for supplementary scrubbing by means of a dilute alkaline solution, if necessary.

(e) The **measurement of smoke density** is a rather intangible process at present, hence it is difficult to establish standards at this stage of the art. The most common method is to compare the appearance of the smoke with the shades of a set of five charts prepared by Ringelmann,[13] of which No. 1 is for 20 per cent density, and the other numbers advance by 20 per cent increments, No. 5 being 100 per cent density. There are also smoke indicating or recording instruments available; in some, the smoke intercepts some of the heat radiated from a uniform source to a thermocouple, in others it affects the intensity of light passing from a lamp to a photoelectric cell. The readings of these measurements usually are given in terms of Ringelmann numbers.

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[11] "Collecting the Dust from Chimney Gases of Powdered-Fuel Installations," by K. Toensfeldt, Trans. A.S.M.E., 1928, FSP 50–59, p. 77.

[12] "An Object Lesson from Battersea," in Power, Nov. 4, 1930, p. 716; also "Removal of Sulphur from Chimney Gases," by David Brownlie, in Combustion, Feb., 1931, p. 45.

[13] A.S.M.E. Code on Instruments and Apparatus, Preliminary Draft of Smoke Density Determinations, in Mechanical Engineering, Nov., 1930, p. 999. Ringelmann charts can be obtained from the Bureau of Mines.

Many communities have ordinances against atmospheric pollution by exit gases, and have bureaus for enforcing them and for devising and giving advice on methods of avoiding pollution.[14]

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[14] See List of Publications of the U. S. Bureau of Mines, for various bulletins, technical reports, serials and circulars on city smoke ordinances and on smoke abatement. Circ. 6262 gives a Proposed Standard Smoke Ordinance, prepared by representatives of the A.S.M.E., A.S.H.V.E., Stoker Mfrs. Assoc., Am. Civic Assoc. and Bureau of Mines.

ADDITIONAL REFERENCES

"The Measurement and Properties of Cinder and Fly-Ash," by A. Stern, Combustion, June 1933, p. 35.

"The Calculation of the Dispersion of Flue Dust and Cinders from Chimneys," by H. O. Croft, Trans. A.S.M.E. 1935, FSP–57–2, p. 5.

"Cinder and Fly-Ash Measurement," by P. H. Hardie, Combustion, March 1935, p. 11.

"Progress in the Removal of Sulphur Compounds from Waste Gases," by H. F. Johnson, Combustion, Aug. 1933, p. 19.

"Removal of Sulphur Compounds from Stack Gases in Great Britain," by D. Brownlie, Combustion, Feb. 1934, p. 18.

"Removal of Sulphur Dioxide in Flue Gas," by C. C. Scott, Combustion, March 1934, p. 29.

CHAPTER XXX

GENERAL FUNDAMENTALS REGARDING STEAM-GENERATING UNITS

392. Introduction.—(a) The developments that have taken place in the equipment and methods used in generating steam have occurred very rapidly in recent years, especially since 1919, and probably have been more pronounced and revolutionary than those made in any other part of the steam-power plant in that time. They have resulted, in many cases, in decided alterations in sizes, shapes, details of construction, and types of apparatus, and have now rendered inadequate descriptions of such apparatus given in any but the more recent publications. The improvements in the equipment, together with the better methods of operation simultaneously developed, have resulted in better thermal performances and in obtaining greater steam-generating capacity in a given space or with a given physical size of apparatus; and such developments are still going on, with many possibilities yet in sight for making still further advancements. These changes have made it necessary to introduce new technical terms and definitions into the literature on the subject, and even to modify the meanings of some of the terms formerly used, while other terms, which were never entirely appropriate or exact, are now being discarded as obsolete.

(b) The purpose of this chapter is, first, to present a very brief description of the more important parts of the apparatus used for generating steam, and to show the relations between these parts; secondly, to define certain of the terms that are used in discussing such apparatus and its performance; and, thirdly, to give the general method of analyzing the results of tests of this kind of equipment. The more detailed discussion of fuel-burning and of the design and application of the more important types of boilers, superheaters, economizers, air preheaters, furnaces and fuel burners, will be given in subsequent chapters.

(c) The **source of the energy** used to generate steam is commonly *fuel* which is burned in a furnace directly for this purpose, but there are various other sources, some of commercial importance. Thus a considerable amount of steam is generated by the *waste heat* recovered from metallurgical and industrial furnaces. Occasionally, a water-power plant can use economically some of its *electrical energy* (particularly during periods of light load) to generate steam that is needed for various industrial applications in the region served. There are a few places in the world where fuel is very expensive but *sunshine* is plentiful, and then it may sometimes be commercially feasible to generate steam in special boilers designed to absorb the solar energy directly. However, the first cost of any sun boiler that has thus far been built is far too great to enable such a unit to be of economic importance except under a very unusual set of conditions. Also, there are a few localities in which there are *hot springs* or *geysers* which afford sources of heat that may be used for generating steam.

But in all such cases it must be carefully noted that the fixed charges on the initial investment for a very expensive plant, even though using no fuel, may make the steam more costly than that obtained by using the heat developed by burning a suitable fuel. This last statement implies that steam-generating equipment is *relatively inexpensive* when utilizing the energy obtained from fuel. In general, this is true because the apparatus is *designed to effect a very rapid rate of heat transmission* and consequently to keep down the extent of heating surface required, thus minimizing the first cost.

Since the same general principles involved in steam generation hold for all cases, and since *fuel* is the principal source of energy for this purpose, the discussion of steam generation will be limited chiefly to units using this source of energy.

393. Steam-Generating Units.—(a) The fuel-burning steam-generating unit consists in general of (1) apparatus required for making available heat from the fuel, and (2) that needed to transfer to the fluid being heated as much of this energy as is commercially feasible. The apparatus must function in such a manner and transmit heat to the working substance at such a rate that the resulting steam generated will be delivered as rapidly as needed and in the desired condition. The steam-generating unit proper consists at least of (1) *a boiler* and (2) a *furnace*; and in addition may include some or all of the other heat-absorbing elements: (3) a *superheater*, (4) an *economizer*, (5) an *air preheater*, (6) a *reheater*, (7) *air-cooled furnace walls*, (8) *water-cooled furnace walls*, (9) *water-screens* or an air- or water-cooled furnace bottom. Besides these main elements, certain auxiliary equipment, "*auxiliaries*," also may be needed to aid in the proper functioning of the unit.

The primary elements of the apparatus will now be described very briefly, and approximately in the historical sequence in which they came into use. (Each of these parts will be discussed more in detail in a subsequent chapter.)

(b) Originally the steam generator consisted primarily of a **boiler** and **furnace,** both usually contained within a brick *setting,* the walls of which were lined with *refractory material* (firebrick) where exposed to the heat of the furnace or gases. The fuel was fired by hand and was burned on *grates* within the furnace. The simplest units used at the present time are similarly arranged, one being illustrated in Fig. 537, in which the more important parts are labeled. The boiler shown consists of a cylindrical shell containing tubes which extend from head to head and through which the furnace gases pass. The boiler is filled with water to a level slightly above the upper tubes, the remaining volume within the shell constituting the *steam-space.* Because the gases pass through its tubes, such a boiler is termed a *fire-tube* or *tubular boiler.* There are many other arrangements of boilers of this type. The form shown in Fig. 537 is called a *return tubular* boiler because the hot gases from the furnace after sweeping backward over the bottom of the shell return forward through the tubes. The steam generated in such a boiler is delivered in a *saturated* condition and usually contains a small percentage of moisture.

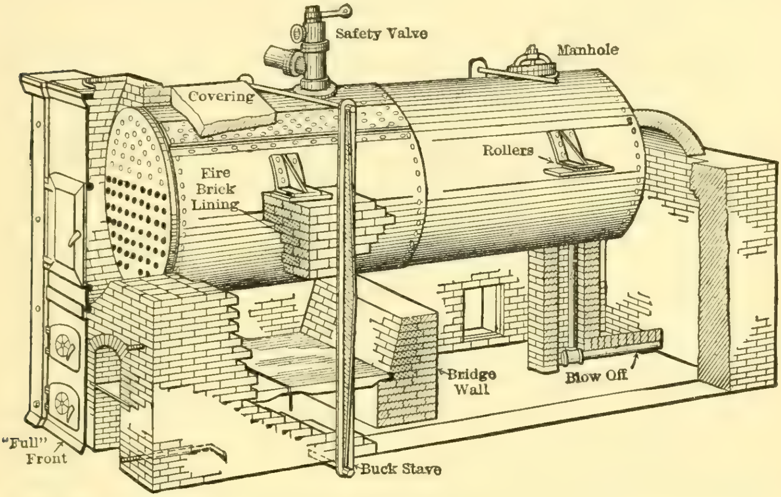

FIG. 537.—Simple Hand-fired Steam-Generating Unit with Horizontal Return Tubular Boiler.

(c) After *superheated* steam was found to have certain advantages for many purposes, later units were often equipped with **superheaters.** One arrangement of this element is shown at S in Fig. 538 and consists of tubes through which the saturated steam from the boiler is made to flow and become heated by the hot furnace gases passing over their external surfaces. Incidentally, the boiler shown in Fig. 538 is one arrangement of the *water-tube* type, i.e., the water is within the tubes, the furnace gases being external thereto.

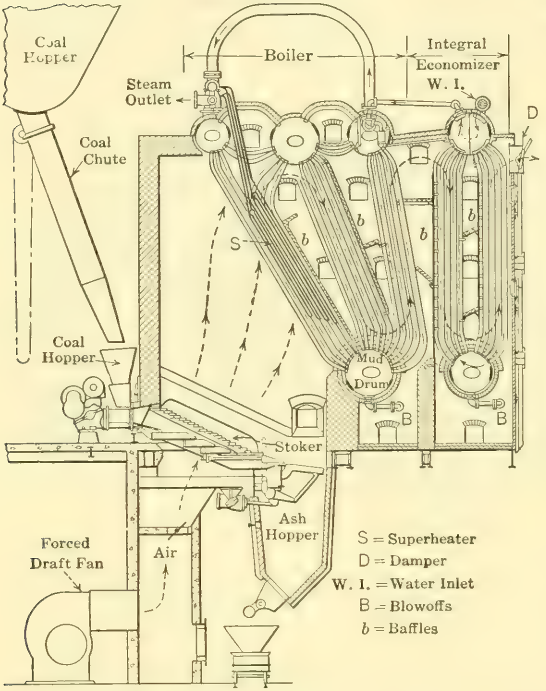

Fig. 538.—A Stoker-fired Steam-Generating Unit Containing a Water-tube Boiler, Superheater, and Economizer.

(**d**) As the products of combustion cannot be cooled in a simple unit to a temperature below that of the saturated water within the boiler, but are always at a much higher temperature in order to have a considerable temperature difference even with respect to the last heating surface swept, much heat is carried to the stack by the gases from such units. To recover some of this heat **economizers** were often added to the simple units. These elements usually consist of tubes which are placed in the path of the exit gases and through which the feedwater is passed on its way to the boiler. The heat thus recovered reduces the amount of fuel required for converting the feedwater into steam and thus increases the efficiency of the unit. Fig. 538 shows one form of economizer,—in this case arranged within the same setting with the boiler, and hence called an *integral economizer*. Other economizers are of the separate or *independent type* and are placed outside of the boiler setting.[1]

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[1] The term *economizer* is now commonly limited to the apparatus which is used to heat only the *feedwater* by means of heat from the boiler exit gases, and *preheater* is generally applied to the apparatus used for heating combustion *air* by energy from the same source. In the transition period such terms as "water preheaters," "fuel economizer (water)," and "air economizer" were often used. Now *air heater* is quite commonly substituted for air preheater.

(e) To reduce the labor required in firing the furnace, to increase the rates of combustion obtainable, and to improve the performance of the unit, **mechanical stokers** were introduced. Labor saving was also effected by installing coal and ash-handling machinery. And to supplement the natural draft of the chimney so as to supply the air at the rates needed for the more rapid combustion, **forced draft fans** or **induced draft fans,** or both, were added. These, like the stoker, were driven by steam engines or electric motors, the energy for which was usually taken from the plant's output.

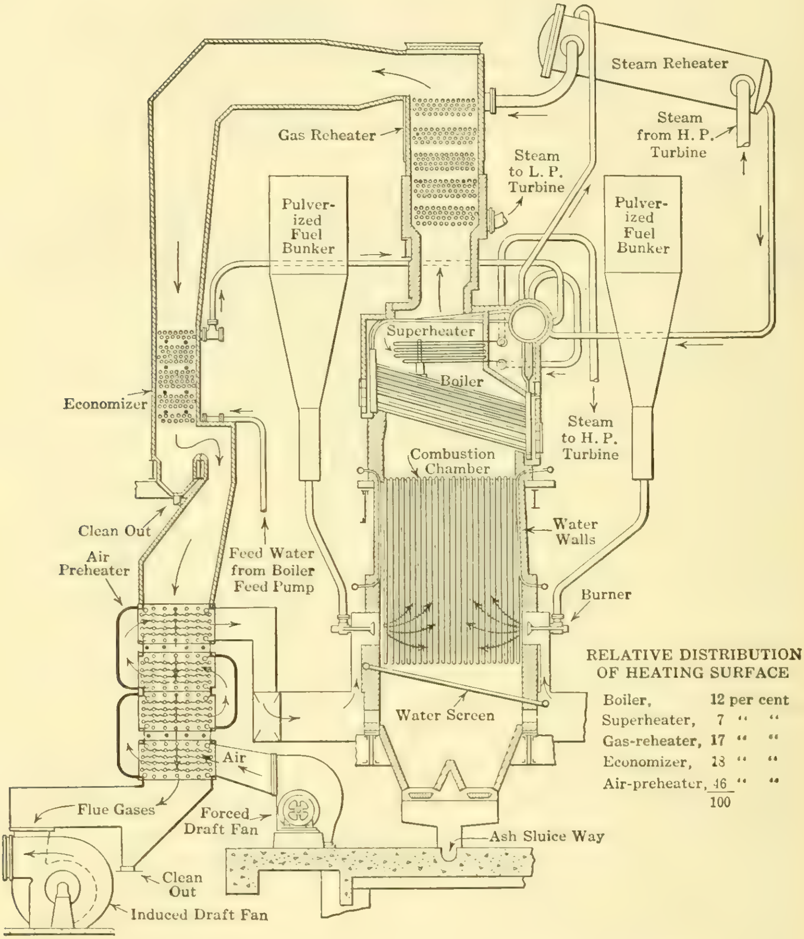
(f) With the introduction of regenerative feedwater heating by means of steam extracted from various stages of the prime mover, the effectiveness of the economizer was reduced, and the gases went to the stack at higher temperature. Hence, **air preheaters** were sometimes added in order to recover some of the heat that would thus be wasted. In such apparatus the air used for combustion is generally heated by contact with walls of tubes or plates the other sides of which are swept by the hot exit gases. The heat returned to the furnace in this way reduces the amount of fuel burned. Fig. 539 shows a unit containing both an economizer and an air preheater, the firing being by means of powdered coal.

(g) With the still higher rates of combustion later used, both with stoker firing and with the more recent *powdered-coal firing*, trouble with refractory walls developed, and this led to using **air-cooled walls** (i.e., the walls were made hollow and were cooled by circulating through them some of the air used for combustion), and later to using **water-cooled walls,** one arrangement of which is shown in Fig. 539.

With powdered-coal firing trouble also arose from the formation in the ash pit of fused masses of cinder. One method of overcoming this difficulty was to place water-cooled pipes, known as **slag screens,** across the bottom of the furnace. These cause the falling cinder particles to be cooled and to deposit in sand-like form which is easily removed. These additions to the unit are shown in Fig. 539. **Water-cooled floors** and other expedients are also used for overcoming this difficulty.

With powdered-coal firing it is also necessary to have such auxiliary **preparative equipment** as dryers, pulverizers and conveyers, which also are strictly parts of the unit, and which consume energy in their operation.

(h) Finally, with the commercial application of the use of high steam pressures in power plants, which makes it necessary to reheat the working substance after it has passed part way through the prime mover, **reheaters** were added, and were commonly incorporated in the boiler units in such a way that some of the heat released in the furnace passes over their tubes through which flows the partly expanded steam which



RELATIVE DISTRIBUTION OF HEATING SURFACE

Boiler,	12 per cent
Superheater,	7 " "
Gas-reheater,	17 " "
Economizer,	18 " "
Air-preheater,	46 " "
	100

Fig. 539.—A Steam-Generating Unit which Includes Powdered-coal Burners, Water-tube Boiler, Superheater, Economizer, Air Preheater, Gas Reheater, Steam Reheater, Water Walls, Water Screens, and Forced and Induced Draft Fans.

is to be heated. When the furnace gases are used for the purpose the element is called briefly a **gas reheater,** although the term is somewhat unsatisfactory. Fig. 539 shows this type of reheater in the flue-gas passage above the boiler.

Reheating can also be accomplished by means of high-temperature steam, in which case the apparatus is termed a **live-steam reheater,** or, more simply, a **steam reheater.** An element of this type is shown in Fig. 539 in series with the gas reheater, and is arranged to receive the live steam directly *from the steam drum* of the boiler to which the drip returns by gravity. In other arrangements of steam reheaters the live steam is obtained from the steam main at a considerable distance from the boiler.

(i) The **fuel-burning equipment** is basically dependent in design and operation on the kind of fuel used, i.e., whether solid, liquid or gaseous. Coal in mass is burned on hand-fired grates or on mechanical stokers; powdered coal and liquid and gaseous fuels are burned in suspension in the combustion air and require special forms of "burners." The details of combustion apparatus will be discussed later.

(j) The **auxiliaries** of a steam-generating unit are those used: (1) to prepare the fuel for burning (e.g., crushers, driers, pulverizers, etc.); (2) to deliver the fuel to the furnace in proper form for mixing with the air (e.g., conveyors and feeders or fuel pumps); (3) to force or draw the air into the furnace (forced or induced draft fans); (4) to pull the gaseous products of combustion through the unit and deliver them to the stack (induced draft fans); (5) to remove the refuse from the ash pit (ash disposal apparatus); (6) to remove fly ash and cinder from the stack gases (cinder catchers or dust and smoke eliminators); and (7) to clean the heating surfaces (soot-blowers for cleaning external surfaces, and blowoffs for removing internal deposits).

The methods of estimating the energy required to operate the auxiliaries of a steam-generating unit will be found in Sect. **278.**

(**k**) The **feed pump** is *not* an auxiliary of a steam-generating unit, for the following reasons: All of the auxiliaries listed in the preceding section are directly involved with either one or the other of the two functions of a steam-generating unit as defined by the 1930 A.S.M.E. Test Code; viz., (1) the development of heat from fuel and (2) the transmission of heat from the products of combustion to another fluid. On the other hand, the feed pump performs its own distinct function, entirely disconnected from either the generation or transmission of heat; it builds up pressure in the working substance which is thereby enabled to enter some element of the steam-generating unit. The entering condition of the fluid is determined by its temperature and pressure; the temperature is dependent upon the external feedwater heaters (if any are used), and the pressure upon the feed pump. Both the heaters and the pump add energy to the feedwater *before* it enters the steam-generating unit; if one is classed as an auxiliary to this unit so should the other be also. The heaters are not so classed and neither should be the pump. And, again, for any rate of flow of steam the energy required to operate the real auxiliaries of this unit is directly dependent upon the cleanliness of its heating surfaces, the kind of fuel used and the operation of the fuel-burning equipment; clearly the feed pump is not in this class of equipment. Thus there are several basic reasons why the feed pump cannot logically be considered as an auxiliary of the steam-generating unit. The energy consumption of a feed pump is treated on page 121.

394. Measuring Heating Surfaces of Steam Generators.—(a) In modern steam generators the heating surfaces consist of such a great variety of kinds and arrangements that the measurement of these surfaces requires careful definitions, if performances are to be compared on a fair or reasonable basis. The following definitions are those given by the 1930 *Test Code for Stationary Steam-Generating Units,* as prepared by a committee of the American Society of Mechanical Engineers.[2]

(b) "The *heating surface of a steam-generating unit* shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas or refractory being cooled and on the other to the fluid being heated, *measured on the side receiving heat.*" Clearly this surface is equal to the sum of (1), (2), (3), (4), and (5) as defined in the succeeding paragraphs.

(1) "**Boiler heating surface** shall consist of that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, *in which the fluid being heated forms part of the circulating system.* This surface shall be *measured on the side receiving heat.* This includes the boiler, water walls, water screens, and water floor. **Heating surface located in the furnace** *or furnace boundaries* shall be measured as follows:

- (A) "Tubes wholly exposed or wholly or partly embedded in refractory—that portion of the surface of the tubes which is exposed to the gas or refractory being cooled.
- (B) "Tubes provided with extended surface—that portion of the surface of the tubes and the extension which are exposed to the gas being cooled.
- (C) "Tubes protected by blocks rigidly attached to the fire faces—that portion of the surface of the blocks which is exposed to the gas being cooled.

(2) "**Superheater surface** shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas being cooled and on the other to steam being heated after leaving the boiler circulating system, the surface being *measured on the flue gas side.*

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[2] The use of bold-faced type and italics has been added by the authors.

(3) "**Economizer surface** shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas being cooled and on the other to fluid being heated before entering the boiler circulating system, the surface being *measured on the flue gas side.*

(4) "**Reheater surface** shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas being cooled and on the other to steam being heated after expansion, the surface being *measured on the flue gas side.*

(5) "**Air heater surface** shall consist of that portion of the surface of the heat-transfer apparatus exposed on one side to the gas being cooled and on the other to air being heated for combustion, the surface being *measured on the flue gas side* except that in air heaters of the regenerative type, where the surfaces are alternately in contact with gas and air, the air heater surface shall be determined for the purpose of this code by multiplying the total surface of the apparatus by that proportion continuously in contact with the gases being cooled."

(**c**) The *heating surface* in a **steam reheater,** used to reheat steam by means of steam having a higher pressure, is *not* included in the measurement of the heating surface of a steam-generating unit, because the surface in this type of reheater does not absorb any heat directly from the products of combustion, but only from steam that has in turn received this energy from the boiler heating surface.

However, when measuring the *performance* of steam-generating units having certain arrangements of steam-reheaters, the *energy* absorbed by the reheated steam in passing through this part of the unit must be considered; and this is done in the manner explained in connection with Eq. (569), page 442.

395. Capacity, Size, Load, and Rating of Steam-Generating Units.—(**a**) Some of the main pieces of equipment in a power plant are operated for the purpose of *transforming energy* from one form to another—such are illustrated by a steam turbine, an electric generator, or a furnace; other equipment merely *transfers* energy from one fluid to another. The steam-generating unit, however, is a combination of apparatus that involves both *transformation* and *transfer of energy.* A chemical *transformation of energy takes place* in the furnace and then *heat is transferred* from the products of combustion through the various heating surfaces of the unit. Even when an apparatus is built solely for the purpose of transforming energy there is not universal agreement as to how its *size* should be expressed. For example, should the *size* of a prime mover be specified in terms of the maximum power it will develop regardless of its efficiency; or should it be given in terms of the *power developed* at maximum efficiency? These questions are not always answered in precisely the same way by various engineers, nor are they the same for different types and classes of prime movers; but in all cases the *rate of energy transformation, or power,* is likely to be the basis of the answer to the question, "How big is this unit"? Thus, for example, one might designate the *size* of a turbine as 5000 hp., and at the next breath point out how much *smaller* the turbine is than the 1000-hp. piston engine which it displaced. Here is clearly a *confusion of physical dimensions and power* as a measure of size. Furthermore, when comparing the *size* of one power plant with another, the custom is to think only in terms of the *capacity* of each plant measured in some convenient *power unit,* rather than the dimensions of the plant building. Thus the term *size* is a rather flexible one, in engineering as well as in general usage, because the standard of measurement is not specified when the word is used. When trying to apply the word *size* to the combination of apparatus designated as the *steam-generating unit* the question becomes still more complicated; even worse is the situation as to the use of the word *rating* when applied to such a unit, because rates of heat transfer per unit area of heating surface and rates of energy transformation per unit volume of furnace (or per unit area of grate) are then involved. The succeeding sections will assist in clarifying the use of these terms and certain others used in connection with steam-generating units.

(**b**) The **size** of any apparatus consisting primarily of *heating surface* may be expressed very logically in terms of the amount of this surface. This applies to the *condenser, feedwater heater, economizer, air preheater, superheater,* and also to the *boiler* when this term refers to its heating surface alone.

When there is an *aggregation* of different heat-absorbing elements in a steam-generating unit, the size of the unit can be shown only by giving the amount of heating surface in each heat-transmitting part of the unit, together with the grate area, or capacity of fuel burner, and the furnace volume; and, in addition, a drawing of the unit may also be needed to show the full significance of the arrangement and effectiveness of the unit's components. Often, instead of giving this information as to physical size, it is merely necessary to give the *capacity* of the unit to generate steam.

(**c**) The **steam-generating capacity** of the unit may be expressed either in terms of

(1) The maximum rate of heat absorption by the unit, expressed either in **kilo B.t.u.** (kB) **per hour,** or in *mega B.t.u.* (mB) *per hour* [3], or

(2) The maximum rate of steam formation, in **pounds per hour,** or multiples thereof, together with a statement of the pressure and temperature of the steam and of the feedwater.

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[3] kB signifies 1000 B.t.u., and this quantity was defined in 1922 by a committee of the A.S.M.E. as the **Unit of Evaporation**; but this term is not recommended in the 1930 Code, nor by the authors. $mB = 1000kB = 1{,}000{,}000$ B.t.u.

Of these two forms of expression the first one is preferred for accuracy, since it gives a clear-cut statement of the rate of energy absorption from the products of combustion. The second one is very short if nothing is said as to the pressures and temperatures involved, and is therefore very convenient, while often sufficiently exact. But, while the weight of steam generated per unit of time is, of course, the most important factor involved, it is essential when accuracy is desired, to include also the statement of these pressures and temperatures, and this is especially important when comparison is to be made between test results from plants in which the steam and feedwater conditions are dissimilar.

While the statement that a unit has generated a maximum of so many thousand or million pounds of steam per hour gives the capacity that has been obtained with a unit, it may not be the maximum capacity that could be safely used. The *real* maximum capacity of a unit is seldom obtained on account of the uncertainty of a cautious operator as to how high a rate of firing the furnace, fuel-burning equipment, or other parts will stand.

(**d**) The **load** of a steam-generating unit at any time is the actual rate of heat absorption or delivery of steam at that time, and is therefore expressed in the same units as capacity. Thus the maximum load that can be carried becomes the "capacity"; and the use of pounds of steam per hour to express the various loads carried during a series of tests is satisfactory if the pressures and temperatures of the steam and feedwater have been held constant, or nearly so, during each test.[4]

If a definite capacity of a steam-generating unit could be specified as its real maximum, then all other loads on this unit could be referred to in terms of this maximum; or if some definite load on the unit could be *easily and accurately* determined as the most efficient one, then other loads could be referred to it as the **"rated"** or **normal load.** If either scheme could be successfully applied, then real information would be given by the statement that the unit is operating at some definite "*per cent of rating*"; but this procedure involves so many difficulties that, at the present time at least, the A.S.M.E. Power Test Code Committee eliminates the word "rating" from its (1930) code for stationary steam-generating units. One of the reasons for discarding the term is that confusion with the *old term, "normal rating,"* would surely arise. The reasons for this will be clear after reading (**e**) of this section which deals with obsolete terms.

One logical method of stating the intensity at which a unit is operating, or the **load intensity,** is to give the *rate of heat absorption per unit time and per unit area*; this may be expressed in B.t.u. per hr. per sq. ft. of total surface; or, instead of giving an overall figure for the unit, separate values for each type of heating surface may be used. The latter method is very expressive but rather complicated, while the overall figure is of doubtful value in comparing two units having entirely different distributions of heating surfaces.

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[4] Often the term "capacity" has also been used as synonymous with "load" as here defined; the greatest output possible then being termed the "maximum capacity."

Since *load* is fundamentally a rate of energy absorption, a very logical procedure would be to express this rate in the familiar power unit, the kilowatt, which is equivalent to about 3413 B.t.u. per hr. Many engineers, including the authors of this text, look with favor on such a method of expressing the load on a steam-generating unit.

(**e**) Some **obsolete terms** pertaining to boilers will now be defined because they have been used in the literature of the past to which references may often need to be made. Furthermore, some of these terms will probably still be used in the commercial world for some years yet because changes require time to become generally effective.

Many years ago boiler manufacturers adopted the rule of expressing the size of the boiler by what they called the "**nominal boiler horse power**" which was arbitrarily taken as represented by 10 sq. ft. of heating surface.[5] Thus a boiler having 5000 sq. ft. of heating surface was designated as a 500-hp. boiler by the so-called "*builder's rating.*" This method is still continued to some extent, especially for small units, because it has become well established in trade catalogues and similar publications, but it is not used in the best engineering literature to-day.

In 1876 the "**boiler horse power**" was defined by a special committee in charge of boiler tests at the Centennial Exposition as the *evaporation of* 30 *lb. of water per hr. from an initial temperature of* 100 *deg. fahr. into steam at* 70 *lb. per sq. in. gage.* This particular rate of evaporation was chosen because it represented the average steam rate of the engines in use at that time. The committee believed that it would be desirable to have the horse power of the boiler agree with that of the engine for which it furnishes steam. This definition was standardized by the American Society of Mechanical Engineers, when it was organized in 1880. It soon became evident that, to take care of the various amounts of superheat, steam pressures, and feedwater temperatures, there was needed a more convenient rate of evaporation for this unit, which, however, was to be equivalent to the original boiler horse power as to the rate of energy absorption. The unit was therefore redefined in terms of the amount of energy required to convert saturated water at 212 deg. fahr. into saturated vapor at the same temperature. On this basis the **boiler horse power** is defined as *the equivalent of evaporating* 34.5 *lb. of water per hour "from and at* 212 *deg. fahr.*"; and the term **unit of equivalent evaporation** was applied to the heat of vaporization of water "from and at 212 deg. fahr." Since the latent heat of steam at 212 deg. fahr. is very close to 970 B.t.u. per lb., the boiler horse power is therefore equivalent to approximately 33,500 B.t.u. per hr.[6] With this rate of heat absorption and with the "*manufacturer's rating*" of 10 sq. ft. of heating surface, each representing the so-called "boiler horse power," consistency requires that the heating surface should absorb energy at the rate of 33,500 10, or 3350 B.t.u. per hr. per sq. ft. This was the so-called "normal rating" of a boiler and it was often expressed as the "equivalent evaporation" of 3.45 lb. of water per hr. from each sq. ft. of surface. For example, if a boiler should absorb heat at the rate of 13,400 B.t.u. per hr. per sq. ft. it would, on this old basis, be said to operate at **"400 per cent of rating."**

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[5] 12 sq. ft. or more was also often used, but 10 has become much more generally adopted in this country.

[6] By Keenan's Table this value is 33,472, and by Goodenough's it is 33,523.

One difficulty encountered in reading the reports of some of the old "boiler tests" arises from the lack of clearness as to the amount of heating surface that was included in the so-called "*boiler surface.*" In many cases this surface did not include the superheater or economizer, yet the feedwater temperature entering the economizer and the steam temperature leaving the superheater were used in determining the heat absorbed. With such a procedure extravagant claims could readily be made for extremely "high ratings" of the so-called "boiler."

The **factor of evaporation** is a discarded term that never had any real justification for existence. This factor was first defined as the ratio of the heat absorbed by each pound of water in passing through "the boiler" under actual conditions to the heat required to evaporate one pound "from and at 212 deg. fahr." When this term was first employed superheaters were seldom used, but when such apparatus became very common, the word "boiler" in this definition was enlarged to include them.

396. Performance of Steam-Generating Units.—(a) The *performance of a unit* that consists of heating surfaces, furnaces, and fuel-burning equipment may naturally be expected to vary somewhat with the various types and arrangements of these components; but generally the *condition* in which each part is maintained is a factor of even greater importance. Thus, for example, if the heating surfaces are not kept reasonably clean, or if the furnace grate permits a large loss of fuel, or if the furnace walls permits excessive quantities of air to leak through them, the performance of the unit cannot be expected to be very satisfactory. The *performance* of the unit is generally expressed in terms of its *efficiency,* or by an *energy balance,* or by a combination of both; sometimes "performance" refers to the portion of time the unit is available for service.[7]

(b) The **efficiency of a steam-generating unit** is, in general, the fraction that shows what portion of the heat supplied by the fuel used in a definite time has been absorbed by the water and steam passing through the unit in the same time. Any convenient unit of time may be used to measure both of these quantities of energy so long as the time unit chosen is the same for each; the hour is very commonly selected.

In the earlier tests the energy consumption of the unit's auxiliaries was not charged against the unit, hence the computations gave a *gross efficiency.* Even now the efficiency obtained on such a basis is often all that is needed. But when a measure of the true performance of the unit is required, this auxiliary energy should be considered, so as to obtain the true or *net efficiency* of the unit. If the rate of energy consumption by the auxiliaries is known the net efficiency can be readily derived from the gross, and vice versa. As the determination of the net efficiency involves all the items to be accounted for in the gross efficiency and others in addition, the discussion will be devoted mainly to the net efficiency.

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[7] See "The Performance of Modern Steam-Generating Units," by Hirshfeld and Moran, Trans. A.S.M.E., 1930, FSP 52–34.

In the most general case, which involves a reheater as part of the generating unit, the **net efficiency** *of a steam-generating unit* becomes

$$e_u = \frac{w_n(h_2 - h_1) + w_r(h_4 - h_3)}{w_f(\text{H.V.})} \quad . \quad . \quad . \quad . \quad . \quad . \quad (569)$$

in which

h_1 = the heat content of the feedwater at the inlet to the first heating element of the unit (e.g., to the economizer when installed, otherwise to the boiler itself), in B.t.u. per pound, as determined by t_1 and p_1. (See p. 264, Part I, for a discussion of the effect of p_1 on h_1.)

h_2 = the heat content of the steam at the outlet from the superheater (or from the boiler if no superheater is installed), in B.t.u. per pound.

h_3 and h_4 = the heat content of the steam at the inlet and outlet, respectively, of the reheater,[8] in B.t.u. per pound.

H.V. = the heating value of the fuel, *as fired,* in B.t.u. per pound.

w_f = weight of fuel fired, in pounds per hour.

w_n = net weight of steam delivered by the superheater, in pounds per hour.
= gross weight — auxiliary steam.

w_r = weight of steam passing through the reheater, in pounds per hour.

The last quantity in the numerator of Eq. (569) becomes zero when a **reheater** is not included in the equipment, or when the only one used is of the *steam*-reheating type and is so located that it receives its live steam from the main discharge from the boiler, in which case this steam is included in the item w_n. However, when a reheater of this type receives its live steam *directly* from the *steam drum* of the boiler, as in Fig. 539, the heat so supplied is not accounted for in the steam issuing through the main boiler outlet, and therefore must be treated as a separate item, as in Eq. (569).

(**c**) The **energy used by the auxiliaries** [listed in Sect. **393** (**j**)] may be consumed by steam engines, steam jets, electric motors or air compressors, or a combination of them; but the basic principle of using this energy in calculating the net efficiency of the unit remains the same. In every case the numerator in Eq. (569) should express the *net*

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[8] When a *steam reheater is in series with a gas reheater,* as shown in Fig. 539, the state 3 then becomes that of the steam entering the steam reheater and the state 4 that of the steam leaving the gas reheater.

amount of energy delivered by the unit. If all the auxiliaries are steam driven, the net weight of steam delivered is, of course, the gross weight minus the steam consumed by the auxiliaries; whereas, if some or all of the auxiliaries are not steam driven, the *equivalent weight* of steam needed to supply the auxiliary energy is calculated and is subtracted from the gross weight. Thus, for auxiliaries driven by electricity from the main generator the **equivalent weight of auxiliary steam,** in pounds per hour, becomes

$$w_x = (F_n)(1 - E_a)\left(\begin{array}{l}\text{Rate of}\\ \text{firing,}\\ \text{tons}\\ \text{per}\\ \text{hour.}\end{array}\right)\left(\begin{array}{l}\text{Auxiliary}\\ \text{energy of the}\\ \text{steam-generating}\\ \text{unit, kw.-hr.}\\ \text{per ton.}\end{array}\right)\left(\begin{array}{l}\text{Actual}\\ \text{evaporation,}\\ \text{pounds of}\\ \text{steam per}\\ \text{pound of fuel}\end{array}\right), \quad . \quad (570a)$$

or

$$w_x = (F_n)(1 - E_a)\left(\begin{array}{l}\text{Auxiliary power of}\\ \text{the steam-generating}\\ \text{unit, in kilowatts.}\end{array}\right)\left(\begin{array}{l}\text{Actual}\\ \text{evaporation,}\\ \text{pounds of}\\ \text{steam per}\\ \text{pound of fuel}\end{array}\right), \quad . \quad . \quad . \quad . \quad (570b)$$

in which

F_n = net fuel rate of the station, in pounds of fuel fired per kw.-hr. sent out from the station.

E_a = fraction of the main turbine-generator gross output required for all of the station auxiliaries, as determined by test, or as calculated by Eq. (381), p. 128.

In Eq. (570*a*) the product of the two items, rate of firing and auxiliary energy, represents the total *auxiliary power* of the steam-generating unit, in kilowatts; and when this *power* is known its use simplifies the equation slightly as indicated by Eq. (570*b*). However, with powdered fuel installations using electrically driven auxiliaries, the form of Eq. (570*a*) will generally be more convenient to use, because the *auxiliary energy* per ton of coal fired is commonly given in the test data.

Clearly by subtracting w_x, as determined by Eq. (570), from the gross weight of steam delivered per hour by the unit, the net weight w_n in Eq. (569) is obtained. The relation between w_x, the *gross efficiency* (e'_u) and the *net efficiency* (e_u) of the steam-generating unit having no reheater is given by the equation,

$$e_u = e'_u(1 - w_x/\text{act. evap. in lb. per hr.}). \quad . \quad . \quad (571)$$

(d) In some cases, especially in the smaller plants, a test of a steam-generating unit cannot be expected to be too elaborate because of the expense involved; hence the energy consumption of the auxiliaries during the test may not be measured or even estimated. In such cases the lack of this information may not be very serious, because it may not be needed for the purposes of the investigation, or because some of the measurements regarding fuel and steam may involve a larger error than that due to the neglect of the auxiliaries. Clearly, under these conditions, the *gross weight of steam* delivered must be used in Eq. (569) instead of the *net* weight, and the result becomes the *gross efficiency*. On the other hand, with large modern units tested with great care, the *auxiliary energy is important* and can be accurately measured, especially when electric motors are used; and then *this information should be included in the test data*, as is being done in the better class of work to-day. This auxiliary energy becomes an appreciably higher portion of the energy absorbed by the water and steam when a unit is forced to a very high rate of operation, as may be noted from some of the test data given in Sect. **397.**

(e) An **example** to illustrate the applications of Eq. (569) and (570) will now be given; consider the following case:

Actual evaporation = 110,000 lb. per hr.
Rate of firing = 6 tons, or 12,000 lb. per hr.
H.V. of fuel = 12,500 B.t.u. per lb.
Net fuel rate of station = 1.38 lb. per kw.-hr.
Auxiliary energy of the steam-generating unit = 62 kw.-hr. per ton.
$h_2 - h_1 = 1220$ B.t.u. per lb of steam.
$E_a = 5$ per cent.

Then, from Eq. (570*a*),

$$w_x = 1.38(1 - .05)(6)(62)(110{,}000/12{,}000) = 4460 \text{ lb. per hr.}$$

Consequently, from Eq. (569), assuming no reheater, the *net efficiency* of the unit becomes

$$e_u = \frac{(110{,}000 - 4460)(1220)}{12{,}000 \times 12{,}500} = 85.7 \text{ per cent,}$$

while the *gross efficiency* would be

$$e'_u = \frac{(110{,}000)(1220)}{12{,}000 \times 12{,}500} = 89.4 \text{ per cent.}$$

These results may be checked from Eq. (571); thus,

$$e_u = 0.894(1-4460/110{,}000) = 0.857.$$

(f) The **losses of energy that occur in a steam-generating unit** and thereby affect its efficiency are the following:

(1) *Loss from the incomplete combustion of the fuel* as evidenced by the presence of:
  (*a*) combustible in the refuse collected in the ash pit,
  (b) combustible, such as solid carbon, carbon monoxide, hydrogen, and hydrocarbons, in the exit gas.

(2) *Loss to the stack not caused by incomplete combustion but due to*:

- (a) the dry exit gases leaving the last heating surface at a higher temperature than that of the atmosphere,
- (b) the water vapor, which accompanies the dry gas, leaving the last heating surface at a higher temperature than that of the air and fuel from which it came.

(3) *Loss by heat transfer from the entire unit to the surrounding air.*

The first two items were discussed in Sect. **389**. The last item includes, in the strict sense, all of the mechanical losses that occur in the auxiliaries of the unit. Thus, consider an induced draft fan with water-cooled bearings: all of the friction losses from these bearings pass to the ambient atmosphere or to the cooling water; hence this loss should be included unless this energy is recovered and is returned to the air used for combustion, or to the feedwater.

The magnitude of each of the losses (1), (2) and (3) will vary considerably, depending very largely upon the load, type of fuel, furnace, extent and arrangement of heating surfaces, and also upon the skill of the operator. The greatest loss is often found in that to the stack, because too much air may be used and thus carry away to the stack far more energy than is necessary. On the other hand, if but little excess air is used the combustion is likely to be incomplete and this is far worse than using a slightly excessive amount of air. If the furnace is forced beyond its proper rate the amount of unburned fuel escaping with the ash and to the stack is likely to become large.

(g) The **energy balance**[9] of a steam-generating unit is a tabulation of the amount of energy usefully absorbed by the water and steam and of the amount lost, as determined by actual test. These quantities are usually expressed as percentages of the heating value of the fuel, as fired. Since the loss by heat transfer is not measured directly, it is usually included in the "*unaccounted-for losses*," which also embrace all losses not specifically calculated and the errors of observation and of calculation. The unaccounted-for losses constitute the quantity which needs to be added to the known losses to make the sum equal the total energy supplied.

Obviously, the energy balance is valuable because it shows the magnitude of each measured loss and therefore indicates, by comparison with other tests, possible improvements in the design or operation of the unit, and because it gives a check on the accuracy of the conduct of the test.

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[9] This has very commonly been called the "**heat balance**" but this term is not as satisfactory as "energy balance" because all of the energy accounted for does not enter or leave the unit in the form of heat.

Before the various items in the energy balance can be calculated, the following information must be obtained by test:

(1) The weight of steam delivered by the superheater during the test; also the weight delivered by the reheater.
(2) The average pressure and temperature of the steam leaving the superheater, of the steam entering and leaving the reheater, and of the feedwater entering the unit.
(3) The weight of fuel used during the test.
(4) A proximate or ultimate analysis (or both) of a fair sample of this fuel, from which the total carbon, hydrogen, ash, moisture, and heating value, as fired, are obtained.
(5) The weight of refuse collected during the test, and in cases of extreme accuracy, the analysis of a fair sample of the refuse.
(6) An analysis of a fair sample of the exit gas leaving the last heating surface.
(7) The temperature of the exit gas leaving the last heating surface.
(8) The relative humidity, pressure, and temperature of the air entering the building housing the unit.
(9) The temperature of the fuel entering the building housing the unit.
(10) The energy used during the test by the auxiliaries of the unit.

Because of the many sources of inaccuracy in determining so many quantities, it is doubtful if the error of the usual energy balance is much less than 2 per cent. In other words, if the heating value of the fuel as fired is 12,500 B.t.u. per lb., the error involved in the balance may be of the order of 250 B.t.u. per lb. of fuel. It is therefore important to make all determinations as accurately as possible.

In considering the energy balance, Figs. 540 and 541 are useful. Fig. 540 is a diagrammatic sketch of the steam-generating unit for which a balance is to be compiled.

Note that fuel, atmospheric air, steam to be reheated, and feedwater, enter; while the main steam, reheated steam, wet gas, and refuse leave the unit. A loss of heat by radiation and convection is also shown. The energy required to operate the auxiliaries is not indicated in the sketch; for in calculating the net efficiency, the steam equivalent of this energy is subtracted from the weight of steam delivered by the unit, as explained in Sect. **396** (**b**).

The energy streams entering and leaving the unit are indicated somewhat quantitatively in Fig. 541.

In the energy balance, the *higher* heating value of the fuel is commonly used in the United States as the basis of figuring all items.[10] This is done on the assumptions that, (1) if the energy [11] stored within the entire unit were the same at the end as at the beginning of the test, (2) if combustion were complete, (3) if the exit gas and refuse were cooled to the temperature of the air and fuel, and (4) if there were no external losses of energy, then the water and steam would receive a quantity of heat equal to the higher heating value of the fuel. This premise is subject to a small error in case the air supplied for combustion is not " saturated," or if the fuel contains large amounts of N_2 and H_2; and this error appears as a small item in the energy unaccounted for.

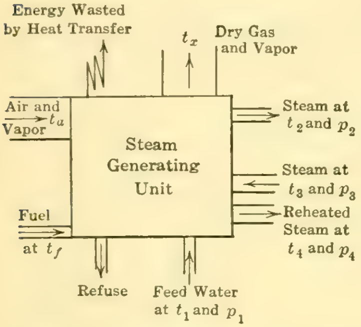

FIG. 540.—Means by Which Energy Enters and Leaves a Steam-Generating Unit.

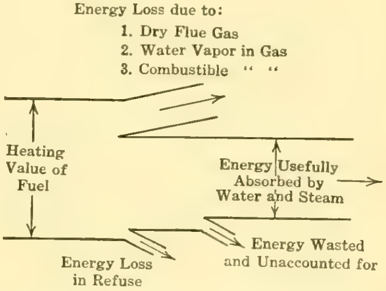

FIG. 541.—Energy Streams Leaving a Steam-Generating Unit.

(h) **A tabular outline of the energy balance** for steam-generating units is given in Table XLVI, and a discussion of the various numbered items follows:

Item (1). In this item, w_{nf} represents the net weight of steam delivered by the unit in pounds *per pound of fuel fired*; and in Eq. (569) the symbol w_n represents the net weight of steam delivered by the unit

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[10] In Europe the *lower* heating value of the fuel is frequently used; hence test results so based are not directly comparable with those reported in this country.

[11] This energy includes that in the working substance and in the fuel bed at the beginning and end of the test. Hence it is important to have the same thickness of fuel bed and the same water level at the beginning and end. The longer the test period the less becomes the magnitude of the error involved in these determinations. Consequently, with stoker- or hand-fired plants the time of test needs to be somewhat longer than when liquid, gaseous, or pulverized fuel is burned, if the same degree of accuracy be desired.

TABLE XLVI

Energy Balance of a Steam-Generating Unit

Item	Method of Determination	B.t.u. per Pound of Fuel as Fired	Per Cent
(1) Energy absorbed by water and steam in unit	$w_{nf}(h_2 - h_1) + w_{rf}(h_4 - h_3)$		
(2) Loss due to incomplete combustion of carbon	$\left(\frac{CO}{CO_2 + CO}\right)C_b \times 10{,}190$		
(3) Loss due to unconsumed carbon in refuse	$w_{cr} \times 14{,}150$		
(4) Loss due to dry exit gas	$0.24w_g(t_x - t_a)$		
(5) Loss due to moisture in fuel	$w_m(1089 + 0.46t_x - t_f)$		
(6) Loss due to H_2O formed from available hydrogen	$9w_{AH}(1089 + 0.46t_x - t_f)$		
(7) Loss due to water vapor in air	$0.46w_vw_A(t_x - t_a)$		
(8) Loss due to imperfect insulation, and unaccounted for	To balance		
(9) Heating value of the fuel, as fired	By test		100.0

C_b = weight of carbon actually burned, in lb. per lb. of fuel as fired.
CO and CO_2 = fractions by volume of these constituents in exit gas.
h_1 = heat content of feed water entering unit, in B.t.u. per lb.
h_2 = heat content of steam leaving unit, in B.t.u. per lb.
h_3 = heat content of steam at inlet of reheater, in B.t.u. per lb.
h_4 = heat content of steam at outlet of reheater, in B.t.u. per lb.
t_a and t_f = temperatures of air and fuel respectively, entering the building in which the unit is located, in deg. fahr.
t_x = temperature of exit gas leaving last heating surface, in deg. fahr.
w_A = weight of dry air used, in lb. per lb. of fuel fired.
w_{AH} = weight of available hydrogen in fuel, in lb. per lb. of fuel, as fired.
$w_c = C_b + w_{cr}$ = total weight of carbon in fuel, in lb. per lb. of fuel, as fired.
w_{cr} = weight of unconsumed carbon in refuse, in lb. per lb. of fuel as fired.
w_g = weight of dry exit gas, in lb. per lb. of fuel fired.
w_m = total weight of moisture in fuel, in lb. per lb. of fuel, as fired.
w_{nf} = net weight of steam delivered by unit, in lb. per lb. of fuel fired.
w_{rf} = weight of steam passing through the reheater, in lb. per lb. of fuel fired.
w_v = weight of water vapor accompanying the air supplied for combustion, in lb. per lb. of dry air.

in *lb. per hour*. Thus $w_{nf} = w_n \div w_f$; similarly, $w_{rf} = w_r \div w_f$. Clearly, if the unit contains no reheater, w_{rf} is zero and this part of item (1) is eliminated. The observed pressures and temperatures determine the values of h_1, h_2, h_3, and h_4.

Item (2). The flue gas analysis is necessary to determine this item. Since carbon monoxide is the *only combustible in the flue gas shown by the usual gas analysis*, the loss due to incomplete combustion, explained in a general way in Sect. **389**, becomes in this case the loss due to incomplete combustion of *carbon to* CO *instead of* CO_2. When one pound of carbon burns to CO instead of CO_2, there is a loss of (14,150 − 3960) or 10,190 B.t.u.[12] Of the total carbon in the fuel given by the ultimate analysis, only a portion is actually consumed in the combustion, and C_b represents this weight of carbon *burned* per pound of fuel fired. Of this carbon that burns, part appears in the CO_2 and part in the CO of the flue gas. The fraction of C_b which burned to CO is $CO \div (CO_2 + CO)$, where the chemical symbols represent the fraction, by *volume*, of these constituents in the flue gas. Thus, the loss due to unburned carbon is clearly as given by item (2) in the table. If there are any other combustibles in the exit gas, such as solid carbon, hydrogen, or hydrocarbons, there is an additional loss due to incomplete combustion which is not calculated either in this or in the usual heat balance. This loss appears in item (8).

Item (3). The refuse collected in the ashpit consists of mineral ash and unconsumed combustible. The exact amount of this combustible and its heating value are difficult to determine. Because of this and also because of the small magnitude of the loss, it is commonly assumed that the difference between the refuse collected per pound of fuel fired and the mineral ash content of that fuel (as shown by the fuel analysis) is unconsumed combustible, which is assumed to be carbon. Since the heating value of carbon is 14,150 B.t.u. per lb., the loss due to unconsumed combustible in the refuse is as shown in item (3). Since the heating value of the combustible in the refuse is uncertain, and also since the refuse carries away some heat by virtue of its temperature, an additional quantity may appear in item (8) as a consequence of the assumptions made in calculating item (3).

Items 4, 5, 6, 7 *and* 8. These have been discussed in Sect. **389**.

(i) The energy balances of a unit tested under various loads may be represented in graphical as well as in tabular form. In such a case, the items of Table XLVI are usually plotted as ordinates *vs.* load, expressed either as the rate of heat absorption, the rate of water evaporation, or, in older tests, the per cent rating, as abscissa.

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[12] See Table XXXVII, p. 368.

397. Some Actual Performance Data.—(a) The previous section has shown how the efficiency and the energy balance of a steam-generating unit may be calculated; a few illustrations of the actual results obtained from some typical units will now be given; additional performance data will also be shown in Chapters XXXI and XXXIII.

(b) **The effect of changing the CO_2** content of the flue gas at various rates of operation of a small unit are shown by the curves[13] in Fig. 542, which were obtained from tests made in rather an unusual manner. The unit was operated in such a way that in each of a series of tests the CO_2 content of the exit gas was held constant at some desired value over a wide range of loads.

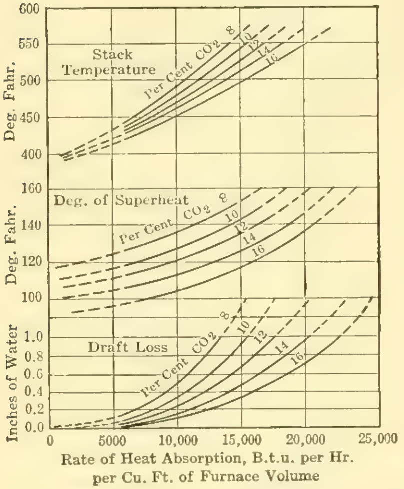

FIG. 542.—Effect of Changing CO_2 and Load.

The curves show that, with any constant value of the CO_2 content, as the rate of operation becomes high the stack temperature, the degree of superheat, and the draft loss are all increased rapidly; but for any constant *load*, as the CO_2 content increases (which means a decrease in the amount of excess air used) the stack temperature, the degree of superheat and the draft loss are all decreased. A high stack temperature means large combustion losses, especially so if much excess air is used, and should not, therefore, be maintained except when the unit must be forced to a high rate of operation.

(c) **Illustrative test results**[14] of a large unit at the Fordson Plant, with and without the air preheater, are shown by the curves in Fig. 543. This unit contained 26,470 sq. ft. of surface in its main boiler tubes, 3014 sq. ft. in its side walls, arches, apron walls and slag screen—or a total of 29,484 sq. ft. in the boiler; the superheater consisted of 3520 sq. ft. in convection surface and 1010 sq. ft. in the side walls of the furnace; the air preheater surface was 39,700 sq. ft.; the furnace was designed for burning both powdered coal and blast-furnace gas and had a combustion chamber of 22,000 cu. ft. The use of the air preheater in this installation clearly increased the efficiency of the unit very appreciably, the amount ranging from 7 to 10 per cent, depending upon the rate of operation. If there had been no preheater the gases would have left the unit at a relatively high temperature as shown by the curve giving the gas temperature leaving the boiler.[15] The curve of carbon loss is seen to increase rapidly as the rate of firing becomes higher; but even at the maximum rate of firing during the test this loss is still not much above 5 per cent. Since the auxiliary energy is not mentioned in the original article, the assumption must be made that the efficiencies given in this figure are gross.

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[13] From data given by Hobbs and Heller in The Proc. of the Engineers' Soc. of Western Pa., Vol. 39, 1923–24. The superheater was of the convection type.

[14] See, "Some Operating Data of Large Steam-Generating Units," by Kreisinger and Purcell, Trans. A.S.M.E., 1928, FSP 50–34, p. 83.

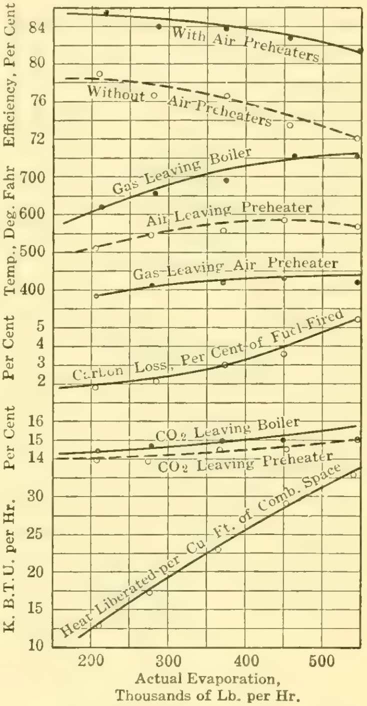

FIG. 543.—Results of Tests of a Steam-Generating Unit, at the Fordson Plant.

(d) **A second illustration** is given by some of the test results of a large unit at Hell Gate, as shown by the curves in Fig. 544, which were prepared from the data given by Grob and Gershberg.[16] The heating surface of the unit consisted of 23,800 sq. ft. of boiler (including 3800 sq. ft. of water walls), 4800 sq. ft. of superheater, 26,136 sq. ft. of economizer, and 11,840 sq. ft. of air preheater; the fuel was powdered coal, which was prepared by the "unit system" and burned with tangential firing.[17] A novel feature of this unit is the *parallel arrangement of the economizer and air preheater,* instead of having them in series as is usual. The former arrangement was chosen partly because of structural limitations imposed by the old building and also to provide satisfactory adjustment of the temperature of the air going to the pulverizer. This adjustment is controlled by the dampers which are normally set to maintain equal gas temperatures at the outlet from the economizer and air preheater; but with wet coal a higher air temperature is needed at the pulverizer, as no coal dryers are used.

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[15] But if no preheater had been installed, the boiler surface would probably have been increased, and possibly an economizer would have been added, in which case the temperature of the exit gas would have been reduced below the values shown in the figure for the gas leaving the boiler.

[16] See Combustion, June, 1931, p. 21.

[17] These terms are explained in Sects. **424** and **426** (d).

The net efficiencies shown in Fig. 544 are obtained from the gross efficiencies by accounting for the loss due to the energy consumed by the coal pulverizers, feeders, forced draft fan and induced draft fan. (Since the original paper gives data to show the energy consumption of all of these electrically driven auxiliaries and also how much the gross efficiency of the unit should be reduced to allow for all but the fans, the total reduction in the efficiency of the unit due to all auxiliaries has been determined by proportion in preparing the curves used in Fig. 544.) The curves show that four burners give better efficiencies than three when the unit is delivering much more than 200,000 lb. of steam per hr.; this is because four burners produce better turbulence within the furnace and four mills grind the coal finer than the three can do at very high loads. The gas temperature from the economizer is seen to rise only a small amount for the range of loads covered by the tests.

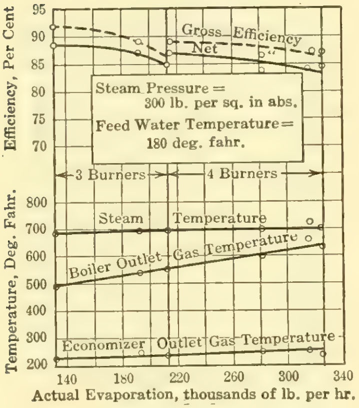

FIG. 544.—Results of Tests of a Steam-Generating Unit at Hell Gate.

(e) A **third illustration** is afforded by the test data [18] of a large unit at the Calumet Station, as given in Table XLVII, in which the most important items are indicated by the bold-faced type. Note that the data in this table were obtained when using *Youghiogheny coal* and a *bin system* [17] of firing pulverized fuel; similar tables are given in the original paper to show the results obtained when using the *unit system* [17] of firing, and also when using other coals with both systems of firing. Regardless of what coal or system of firing was used, the heating surfaces were identical in all cases and were distributed as follows:

Boiler	5,940 sq. ft. or	9.6 per cent
Furnace walls and floor	2,468	4.0
Convection superheater	3,000	4.8
Economizer	8,850	14.3
Air preheater	41,700	67.3
Total	61,958 sq. ft.,	100.0 per cent

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[18] See Paper by A. E. Grunert, Trans. A.S.M.E., 1931, FSP-53-4, p. 43.

The general arrangement of these heating surfaces is indicated in Fig. 545, which also shows the "*unit* system" of firing pulverized coal.[19] The volume of the furnace in all cases was 12,397 cu. ft.

Note that the auxiliary energy, as given in Table XLVII, is on the basis of the *dry ton;* hence, in using Eq. (570) to find w_x, the rate of firing must be given in tons of *dry* coal per hour. Furthermore, instead of obtaining F_n and E_a from this table to use in this equation, the equivalent value of the product of the first two parenthetical quantities in Eq. (570) may be found by dividing 17,000 by the heating value of the fuel as fired. This is because the station was operated under

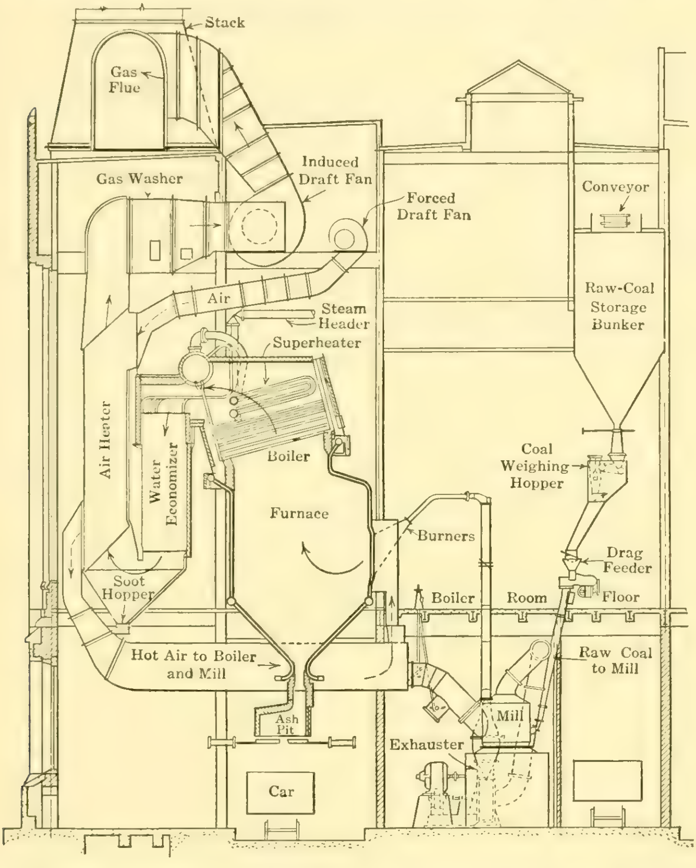

FIG. 545.—Steam-Generating Unit No. 22, Calumet Station, when Using the Unit System of Firing.

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[19] An illustration of a "bin" or "storage system" of firing pulverized fuel (for a different installation) is given in Fig. 539.

TABLE XLVII

EVAPORATIVE TESTS ON UNIT NO. 22, CALUMET STATION, WITH A BIN SYSTEM OF FIRING, USING YOUGHIOGHENY COAL

Test number	1	2	3	4	5	6	7	8	9	10
Date of test, 1928	Aug. 2	Aug. 7	Aug. 9	Aug. 14	Aug. 16	Aug. 21	Aug. 23	Aug. 30	Sept. 5	Sept. 20
Duration, hours	10	12	12	9	12	12	8	8	8	7
Coal as fired:										
Moisture, per cent	2.52	2.63	2.30	2.24	2.21	2.12	2.29	2.22	2.43	2.41
Ash, per cent	12.30	12.84	12.10	12.70	12.70	12.36	12.59	12.53	12.80	11.74
Heating value B.t.u. per pound	**12,589**	**12,426**	**12,625**	**12,601**	**12,558**	**12,651**	**12,568**	**12,563**	**12,479**	**12,693**
Ultimate analysis of fuel, per cent:										
Carbon	83.25	83.26	83.51	83.27	83.23	83.28	83.31	83.30	83.22	83.37
Hydrogen	5.17	5.17	5.18	5.17	5.17	5.17	5.17	5.17	5.17	5.18
Sulphur	2.48	2.47	2.19	2.46	2.50	2.45	2.41	2.42	2.52	2.34
Nitrogen	1.63	1.63	1.63	1.63	1.63	1.63	1.63	1.63	1.63	1.63
Oxygen	7.47	7.47	7.49	7.47	7.47	7.47	7.48	7.48	7.46	7.48
Size of coal (Tyler screens), per cent:										
Through 28-mesh	99.9	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Through 48-mesh	98.3	98.4	98.7	99.0	99.0	99.1	98.5	98.7	99.0	98.5
Through 100-mesh	82.6	93.8	86.2	87.3	87.8	87.6	85.8	85.0	86.5	85.1
Through 150-mesh	75.0	74.8	72.8	79.1	79.6	79.8	77.1	76.3	78.6	76.0
Through 200-mesh	60.0	64.7	68.6	68.5	66.9	64.2	63.8	65.8	69.5	66.5
Feeders in service	1, 2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4	2 & 3	1, 2, 3 & 4	2 & 3	1, 2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4
Firing control	Auto.	Auto.	Auto.	Auto.	Auto.	Man.	Auto.	Auto.	Auto.	Man.
Coal burned per hour, pounds	**16,567**	**19,864**	**16,056**	**11,977**	**16,133**	**9,198**	**16,331**	**16,255**	**23,679**	
Heat liberated per cubic foot furnace volume, B.t.u. per hour	**16,800**	**19,900**	**16,330**	**12,170**	**16,350**	**9,380**	**16,540**	**16,470**	**28,850**	
Water evaporated per hour, pounds	**153,631**	**180,124**	**148,061**	**116,042**	**149,025**	**87,666**	**151,539**	**150,878**	**212,107**	**247,805**
Actual evaporation per pound of coal, pounds	9.27	9.07	9.23	9.68	9.24	9.53	9.28	9.28	8.96	9.50
Total equivalent evaporation per pound of coal, pounds	11.49	11.28	11.37	11.67	11.56	11.38	11.47	11.45	11.30	11.84
Temperature, degrees Fahrenheit:										
Air to **preheater**	111	109	108	93	113	86	98	80	90	91
Air leaving preheater	338	342	324	277	326	259	326	313	368	377
Primary air	113	111	111	112	113	103	107	102	112	107
Secondary air	296	321	306	257	304	235	306	394	342	350
Air to dryer	200	202	191	195	193	194	188	201	188	194
Air leaving dryer	126	124	124	125	126	121	124	120	123	126
Gases leaving boiler	1021	1034	984	838	945	753	975	955	1122	1151
Gases leaving economizer	441	452	411	347	405	305	412	401	509	530
Gases leaving preheater	252	260	239	197	239	187	238	221	289	293
Feedwater to **economizer**	**188**	**189**	**190**	**188**	**190**	**187**	**189**	**189**	**187**	**190**
Superheated steam	**688**	**699**	**682**	**630**	**676**	**610**	**686**	**682**	**727**	**707**
Pressure, superheated steam, pounds per square inch abs	**341**	**338**	**337**	**341**	**340**	**337**	**339**	**341**	**337**	**339**
Pressures, air, inches of water:										
Primary-fan discharge	11.54	11.40	11.50	11.65	11.55	11.76	11.58	11.70	11.54	11.51
Burners in service	4.00	4.90	3.95	5.00	3.90	3.45	4.08	4.01	6.02	6.00
Secondary forced-draft-fan discharge	2.92	3.30	2.40	1.73	2.10	0.60	2.65	2.64	4.06	3.96
Main duct	1.83	1.80	1.40	1.07	1.13	0.14	1.56	1.66	1.95	1.54
Burners in service	0.26	0.59	0.28	0.26	0.12	−0.15	0.23	0.20	1.15	1.53

Drafts; inches of water:										
Furnace	0.07	0.08	0.08	0.07	0.08	0.09	0.08	0.08	0.08	1.0
Uptake	0.70	1.02	0.70	0.40	0.65	0.32	0.75	0.70	1.54	1.85
Economizer out	3.06	4.40	3.00	1.60	2.76	1.10	3.30	3.08	6.78	7.95
Preheater out	3.94	5.70	3.70	1.90	3.50	1.17	4.15	3.89	8.85	10.12
Induced-draft fan inlet (after washer)	4.33	6.20	4.10	2.05	3.83	1.30	4.56	4.22	9.62	11.00
Analysis of gases, per cent by volume:										
CO_2 boiler exit	15.17	14.92	14.68	14.42	14.83	14.17	14.20	14.59	14.93	15.50
CO_2 economizer exit	11.02	13.24	13.66	14.32	14.55	13.95	13.80	13.93	13.90	14.25
CO_2 preheater exit	13.33	12.46	13.01	13.42	13.61	12.76	13.02	12.89	12.90	13.92
O_2 preheater exit	6.16	6.59	6.12	5.85	5.74	6.53	6.28	6.35	6.28	4.85
CO preheater exit										
Ashpit analysis:										
Ash, per cent	98.42	98.75	98.64	97.90	98.03	98.38	99.30	98.62	94.92	98.01
H.V., B.t.u. per pound	229.0	181.0	197.0	305.0	286.0	235.0	102.0	200.0	737.0	289.0
Flue-dust analysis:										
Ash, per cent	97.3	93.65	96.05	94.76	95.12	88.1	94.37	91.08	93.81	93.17
H.V., B.t.u. per pound	384.0	921.0	573.0	760.0	708.0	1717.0	816.0	1293.0	898.0	990.0
Composite (ashpit 40%, flue dust 60%):										
Ash, per cent	97.75	95.69	97.28	96.02	96.28	92.25	96.34	94.10	94.25	95.11
H.V., B.t.u. per pound	322.0	625.0	423.0	578.0	539.0	1124.0	531.0	856.0	834.0	710.0
Energy balance, per cent:										
Absorbed by boiler, economizer and superheater, or **gross efficiency** of the unit	**88.90**	**88.20**	**87.70**	**90.00**	**88.00**	**87.30**	**88.75**	**88.60**	**87.90**	
Loss in dry flue gases	3.74	4.35	3.61	2.65	3.39	2.60	3.74	3.52	5.20	4.99
Loss due to preheated air for drier	0.76	0.67	0.74	1.11	0.74	1.21	0.69	0.64	0.55	0.54
Loss due to moisture in coal	0.22	0.23	0.20	0.19	0.19	0.18	0.20	0.20	0.22	0.22
Loss due to H_2 in coal	3.47	3.52	3.49	3.42	3.47	3.42	3.50	3.50	3.60	3.61
Loss due to unburned carbon	0.32	0.67	0.42	0.61	0.57	1.19	0.55	0.90	0.91	0.69
Loss due to carbon in CO										
Unaccounted for losses	2.59	2.36	3.84	2.02	3.64	4.10	2.57	2.64	1.62	
Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	
Heat transferred by preheater, per cent	4.67	5.02	4.37	3.67	3.90	2.74	4.35	4.39	5.37	
Heat returned to furnace, per cent	3.94	4.35	3.64	2.46	3.16	1.53	3.58	3.75	4.82	
Auxiliary energy, kilowatt-hours per ton of dry coal:										
(1) Feeders	0.37	0.32	0.39	0.40	0.38	0.50	0.37	0.37	0.28	0.27
(2) Primary air	2.62	2.29	2.73	3.03	2.68	3.80	2.71	2.72	2.03	1.78
(3) Induced-draft fan	18.70	29.10	19.10	23.60	18.90	27.70	18.76	19.16	28.00	25.00
(4) Forced-draft fan	6.58	8.10	6.75	9.20	6.73	10.93	6.72	6.99	9.32	8.48
Total of (1), (2), (3) and (4)	28.27	39.81	28.97	36.23	28.69	42.93	28.56	29.24	39.63	35.53
(5) Coal preparation:*										
(*a*) Pulverizer	12.90	12.90	12.90	12.90	12.90	12.90	12.90	12.90	12.90	12.90
(*b*) Exhauster	4.80	4.80	4.80	4.80	4.80	4.80	4.80	4.80	4.80	4.80
(*c*) Fuller Kinyon pump	0.74	0.74	0.74	0.74	0.74	0.74	0.74	0.74	0.74	0.74
Total of (*a*), (*b*) and (*c*)	18.44	18.44	18.44	18.44	18.44	18.44	18.44	18.44	18.44	18.44
Total auxiliary energy of the unit	**46.71**	**58.25**	**47.41**	**54.67**	**47.13**	**61.37**	**47.00**	**47.68**	**58.07**	**53.97**
Heat equivalent of auxiliary energy, B.t.u. per pound of coal as fired	387	482	394	453	392	512	391	397	481	448
Heat equivalent of auxiliary energy, per cent	3.08	3.88	3.12	3.60	3.12	4.04	3.11	3.16	3.85	3.53
Net efficiency of the unit, per cent	**86.20**	**84.70**	**84.90**	**86.80**	**85.30**	**83.80**	**86.10**	**85.80**	**84.50**	

* Prorated figures based on total figures of two months' operation.

such conditions that each kw-hr. of electrical energy used by the auxiliaries required 17,000 B.t.u. from the coal burned by the unit. With this information, the values of the net and gross efficiencies of the unit as given in the table may be checked by using Eqs. (569) and (570).

The net and gross efficiencies obtained from Unit No. 22 with *Kentucky and Illinois* coals are shown in Fig. 546, two systems of firing being used with the latter coal. In this case the bin system of using pulverized coal yielded slightly higher values than the unit system, but with other arrangements the results might be different. The steep drop in the net efficiency that occurs when the evaporation becomes 170,000 lb. per hour is chiefly due to the extra energy required to operate the induced and forced draft fans, additional ones probably being brought into service at this load.

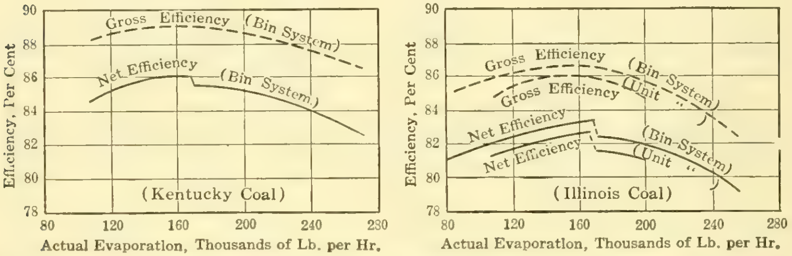

Fig. 546.—Gross and Net Efficiencies of the Calumet Generating Unit with Kentucky and Illinois Coal, the Latter with Two Systems of Firing.

(f) **A fourth illustration** of results obtained from tests of steam generators is afforded by the data from a very comprehensive test conducted on a new unit installed in the Hudson Avenue Station of the Brooklyn Edison Co.[20]. In this test special effort was made to determine accurately the losses given as items (2) to (7) in Table XLVI, thereby minimizing the unaccounted-for loss (8). The test data obtained are given in Table XLVII *a*, and some of the results are shown by the curves in Figs. 546 *a* to *d* inclusive.

The steam generator tested is designated as Boiler No. 74, and has the same arrangement as Unit No. 7 in Fig. 546*e*. It is fired by means of an underfeed stoker, therein differing from the other examples given in this chapter, all of which use pulverized-coal burners. In addition to the boiler, the unit contains a superheater, water walls, and an econo-

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[20] See "The Test Performance of Hudson Avenue's Most Recent Steam-Generating Units," by P. H. Hardie and W. S. Cooper, Trans. A.S.M.E., 1934, FSP-56-15, p. 833.

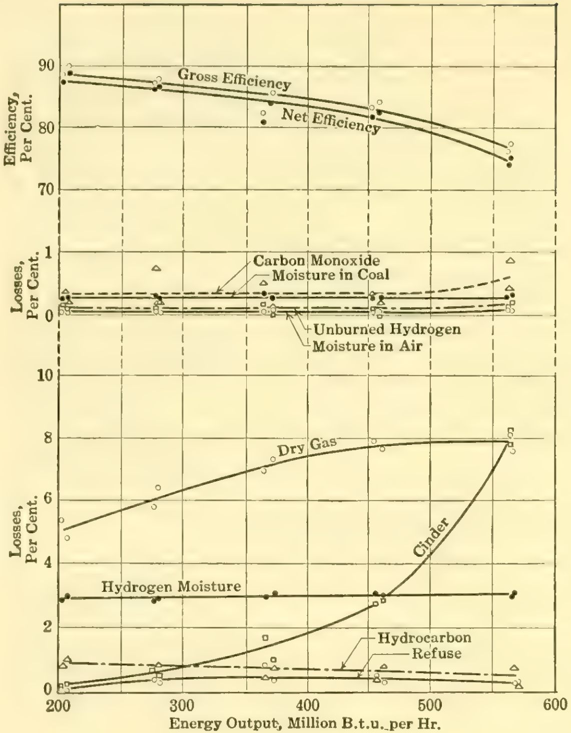

FIG. 546*a*.—Efficiencies and Energy Balance.

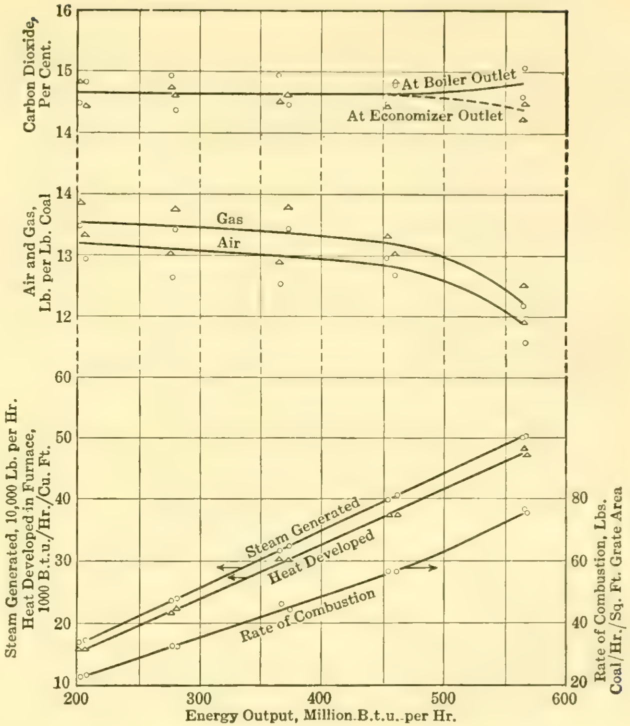

FIG. 546*b*.—Coal, Steam, Air, and Gas Quantities.

TABLE XLVII*a* TESTS OF STEAM-GENERATING UNIT NO. 74, HUDSON AVENUE STATION, WITH STOKER FIRING

Item											
1	Run No.	1	2	3	4	5	6	7	8	9	10
2	Date, 1934	4–2	4–3	4–4	4–5	4–6	4–7	4–8	4–9	4–25	4–26
3	Duration of run, hr	24	24	24	24	24	24	24	24	24	24
	Load										
4	**Energy output, million Btu per hr**	**207.0**	**277.0**	**364.8**	**454.3**	**460.3**	**372.0**	**279.8**	**202.8**	**566.0**	**567.2**
5	**Steam generated, thousand lb per hr**	**174.4**	**237.3**	**318.1**	**399.1**	**405.8**	**324.5**	**240.3**	**171.9**	**501.3**	**504.2**
6	**Coal fired, thousand lb per hr**	**16.50**	**22.61**	**31.76**	**38.92**	**38.87**	**30.69**	**22.68**	**16.32**	**52.83**	**52.24**
	Fuel, Ultimate Analysis (as Fired)										
7	Volatile matter, per cent	18.7	18.4	16.9	15.6	16.2	15.8	16.2	16.3	20.6	21.2
8	Fixed carbon, per cent	70.4	71.3	72.2	73.7	73.9	74.4	73.7	73.6	69.1	68.4
9	Ash, per cent	7.6	6.6	7.2	7.7	6.8	6.6	7.0	7.0	6.9	6.9
10	Moisture, per cent	3.3	3.7	3.7	3.0	3.1	3.2	3.1	3.1	3.4	3.5
11	**Heating value, Btu per lb**	**13,940**	**14,060**	**13,920**	**13,990**	**14,050**	**14,120**	**14,050**	**14,020**	**14,060**	**14,000**
	Fuel, Proximate Analysis (as Fired)										
12	Carbon, per cent	78.7	79.8	79.3	79.7	79.8	80.6	79.5	80.2	80.6	80.0
13	Hydrogen, per cent	4.3	4.2	4.0	4.1	4.0	4.2	4.2	4.2	3.9	4.0
14	Oxygen, per cent	3.6	3.4	3.3	2.9	3.8	2.9	3.9	3.2	2.7	2.8
15	Nitrogen, per cent	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3
16	Sulphur, per cent	1.2	1.0	1.2	1.3	1.2	1.2	1.0	1.0	1.2	1.5
	Flue Gas, Boiler Outlet										
17	CO_2, per cent	14.8	15.0	14.9	14.5	14.8	14.5	14.4	14.5	14.6	15.1
18	O_2, per cent	3.7	3.4	3.6	4.0	3.7	4.0	4.1	4.0	3.8	3.4
19	CO, per cent	0.05	0.20	0.13	0.09	0.05	0.03	0.05	0.08	0.12	0.26
20	H_2, per cent	0.03	0.03	0.04	0.03	0	0	0.05	0.04	0.04	0.06
21	Hydrocarbons, per cent	0.09	0.21	0.04	0.03	0.07	0.06	0.07	0.07	0.07	0.02
22	N_2, per cent	81.3	81.2	81.3	81.4	81.4	81.4	81.3	81.3	81.4	81.2
23	Dry gas, lb per lb coal	13.3	13.0	12.9	13.3	13.0	13.8	13.8	13.9	12.5	11.9
24	Wet gas, lb per lb coal	13.8	13.5	13.4	13.8	13.5	14.3	14.2	14.3	12.9	12.3
25	Dry air, lb per lb coal	13.0	12.7	12.6	13.0	12.7	13.5	13.4	13.5	12.2	11.6
26	Dry air required for ideal combustion, lb per lb coal	10.4	10.5	10.4	10.5	10.4	10.7	10.5	10.6	10.6	10.5
27	Dilution coefficient	1.24	1.20	1.21	1.23	1.22	1.26	1.28	1.28	1.16	1.10
	Flue Gas, Economizer Outlet										
28	CO_2, per cent	14.4	14.7	14.5	14.5	14.8	14.7	14.6	14.8	14.3	14.5
29	O_2, per cent	3.6	3.2	3.4	3.6	3.5	3.8	3.8	3.6	4.3	4.0
30	CO, per cent	0.05	0.08	0.04	0.03	0.03	0.02	0.04	0.07	0.04	0.06
31	N_2, per cent	82.0	82.0	82.1	81.9	81.7	81.5	81.6	81.5	81.4	81.4
	Pressures and Drafts										
32	Moisture in steam leaving drum, per cent	0.7	0.7	0.8	0.8	0.8	0.8	0.8	0.8	0.7	0.7
33	Steam pressure in drum, lb per sq in. abs	426	430	436	447	447	438	429	422	466	466
34	**Steam pressure of superheater outlet, lb per sq in. abs**	**421**	**422**	**425**	**428**	**428**	**426**	**423**	**420**	**431**	**431**
35	Pressure drop in superheater, lb per sq in.	5	8	11	19	19	12	6	2	35	35
36	Pressure drop in economizer, lb per sq in.	13	19	31	45	45	30	17	10	66	68
37	Air pressure in stoker plenum, in. water	1.8	3.0	3.5	4.4	4.2	3.6	3.1	1.6	5.7	6.0
38	Air pressure under extension grate, in. water	0.7	1.4	1.8	2.7	2.3	2.2	2.0	1.4	3.4	3.5
39	Air pressure under ashpit, in. water	0.3	0.5	1.0	1.9	1.2	0.9	0.7	0.4	1.7	1.4
40	Draft in furnace, in. water	0.10	0.11	0.30	0.53	0.59	0.24	0.19	0.07	0.88	1.03
41	Draft at boiler outlet, in. water	0.7	1.4	2.9	4.7	4.7	2.7	1.3	0.6	7.9	7.8
42	Draft at economizer outlet, in. water	1.1	2.0	4.3	7.1	7.1	4.3	2.1	0.9	11.9	11.7
43	Draft at cinder catcher outlet, in. water	1.2	2.2	4.8	7.9	7.9	4.8	2.3	1.0	14.3	14.3

	Temperature										
44	Air temperature, F	64	58	52	53	50	54	55	60	55	60
45	Relative humidity of air, per cent	46	48	53	40	46	53	42	51	48	53
46	Coal temperature, F	66	63	70	76	76	83	80	94	73	70
47	Flue-gas temperature at boiler outlet, F	594	657	713	768	767	728	668	613	817	812
48	Flue-gas temperature at economizer outlet, F	274	317	358	389	386	360	323	287	425	421
49	**Feedwater temperature entering economizer, F**	**212**	**232**	**249**	**263**	**263**	**253**	**232**	**212**	**273**	**274**
50	Feedwater temperature leaving economizer, F	318	345	366	386	384	371	348	321	......	399
51	**Steam temperature at superheater outlet, F**	**711**	**711**	**706**	**716**	**708**	**714**	**706**	**697**	**719**	**712**
	Refuse and Cinders										
52	Combustible in ashpit refuse, per cent	0.6	5.0	10.2	4.8	3.9	4.2	4.0	2.3	3.4	3.7
53	Cinder concentration, lb cinders per thousand lb dry gas	0.59	1.25	2.37	3.15	3.20	1.40	0.85	0.45	7.63	8.32
54	Cinders per thousand lb of coal, lb	7.8	16.3	30.6	41.9	41.8	19.3	11.7	6.2	95.4	98.9
55	Combustible in cinders, per cent	35.7	43.6	54.6	64.0	67.3	53.0	44.1	37.9	79.0	80.9
56	Heating value of cinders, Btu per lb	4990	6090	7750	9140	9730	7550	5960	5120	11,520	11,710
57	Carbon burned, lb per lb of coal	0.779	0.777	0.766	0.765	0.764	0.790	0.783	0.794	0.725	0.716
	Unit Quantities										
58	**Rate of combustion, lb coal per hr per sq ft of grate area**	**23.8**	**32.6**	**45.8**	**56.1**	**56.0**	**44.2**	**32.7**	**23.5**	**76.2**	**75.3**
59	**Rate of energy development in furnace, thousand Btu per hr per cu ft**	**16.2**	**21.7**	**30.4**	**37.4**	**37.4**	**30.3**	**22.3**	**16.2**	**48.0**	**47.2**
60	Steam-fuel ratio	10.6	10.5	10.0	10.3	10.4	10.6	10.6	10.5	9.5	9.7
61	Rate of steam generation, lb per hr per sq ft of surface of boiler & water wall *	6.1	8.4	11.2	14.1	14.3	11.5	8.5	6.1	17.7	17.8
62	Rate of energy transfer, Btu per hr per sq ft of surface of boiler & water wall *	5610	7400	9660	11,820	12,050	9790	7460	5510	14,630	14,700
63	Rate of energy transfer, Btu per hr per sq ft of superheater surface	5120	6970	9230	11,920	11,830	9640	6970	4810	15,070	14,830
64	Rate of energy transfer, Btu per hr per sq ft of economizer surface	840	1230	1710	2290	2290	1770	1280	850	2920	2960
65	Increase in enthalpy in steam-generating unit, Btu. per lb	1187.0	1167.4	1146.7	1138.3	1134.3	1146.5	1164.3	1179.5	1129.1	1125.0
	Efficiency and Energy Balance										
66	**Gross Efficiency of steam-generating unit, per cent**	**90.0**	**87.1**	**82.5**	**83.4**	**84.3**	**85.9**	**87.8**	**88.6**	**76.2**	**77.5**
66a	**Net efficiency of unit, per cent**	**88.8**	**86.1**	**81.1**	**82.0**	**82.8**	**84.2**	**86.6**	**87.3**	**74.4**	**75.7**
66b	Energy required for auxiliaries, per cent	1.2	1.0	1.4	1.4	1.5	1.7	1.2	1.3	1.8	1.8
67	Energy loss due to moisture in air, per cent	0.1	0.1	0.1	0	0	0.1	0	0.1	0.1	0.1
68	Energy loss due to moisture in coal, per cent	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.2	0.3	0.3
69	Energy loss due to moisture formed by burning of H_2, per cent	3.0	2.9	3.0	3.1	3.0	3.1	3.0	2.9	2.9	3.1
70	Energy loss due to incomplete combustion of carbon (CO), per cent	0.2	0.7	0.5	0.3	0.2	0.1	0.2	0.3	0.4	0.9
71	Energy loss due to unburned hydrogen, per cent	0.1	0.1	0.2	0.1	0	0	0.2	0.2	0.1	0.2
72	Energy loss due to unburned hydrocarbons, per cent	1.1	2.4	0.5	0.4	0.8	0.7	0.9	0.9	0.8	0.2
73	Energy loss due to combustible in ashpit refuse, per cent	0	0.4	0.8	0.4	0.3	0.3	0.3	0.2	0.3	0.3
74	Energy loss due to combustible in cinders, per cent	0.3	0.7	1.7	2.7	2.9	1.0	0.5	0.2	7.8	8.3
75	Energy loss due to dry gas, per cent	4.9	5.9	6.9	7.9	7.6	7.3	6.4	5.5	8.1	7.6
76	Energy loss due to radiation, and unaccounted for, per cent	0	−0.6	3.5	1.4	0.6	1.2	0.4	0.9	3.0	1.5
	Power Consumption of Auxiliaries										
77	Stoker and clinker grinder, kw	10	12	13	14	14	12	11	9	19	19
78	Forced-draft fan, kw	188	203	281	297	297	288	228	210	331	338
79	Induced-draft fans, kw	33	78	284	408	399	313	101	37	926	928
80	Total, kw	231	293	578	719	710	613	340	256	1276	1285

* Boiler and water-wall surface = 24,450 + 3846 = 28,296 sq. ft.

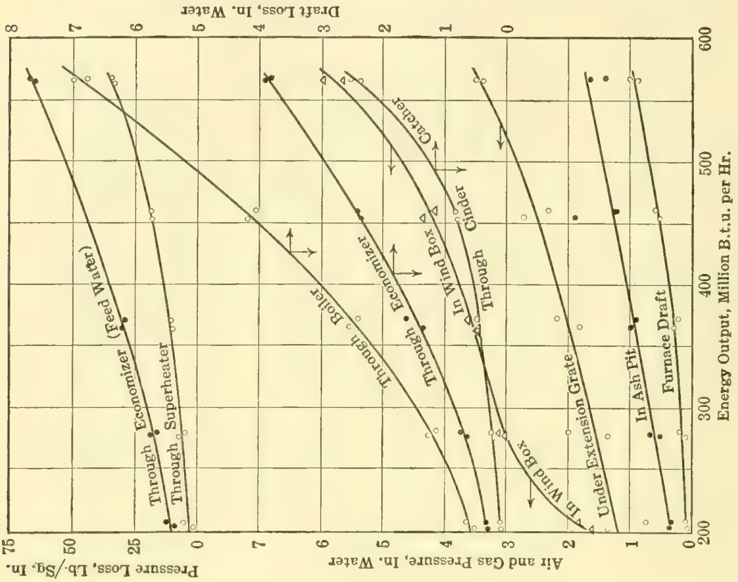

Fig. 546*d*.—Pressures and Drafts.

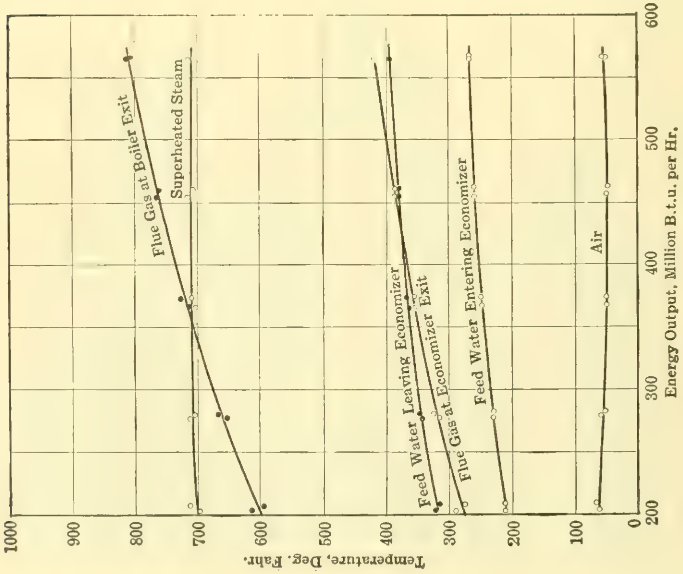

Fig. 546*c*.—Temperatures.

mizer, and it does not use preheated air. The area of the heat-transmitting surfaces in the different parts of the unit are as follows, expressed in sq. ft.: Boiler, 24,450; furnace water walls, 3846; superheater, 5740; and economizer, 22,400. The furnace volume is 14,000 cu. ft. The stoker is the longest one (26′7″) of its type so far constructed, there being 69 tuyeres and a final overfeed section in its length, and it has 15 retorts. The projected grate area is 694 sq. ft., and the wind box under the grate is equipped with zoned air control, which permits manual regulation of the rate of air supply to the various sections to the grate. The unit is served by one forced-draft fan of 200,000 cfm capacity and two induced-draft fans, each of 155,000 cfm capacity.

It is interesting to note that the unit successfully withstood operation for 48 hours with the enormous combustion rate of over 75 lb. of coal per sq. ft. of grate area, meanwhile generating over a half million lb. of steam per hour with a net efficiency of about 75 per cent. The unit has also been operated at much higher rates over short periods of time.

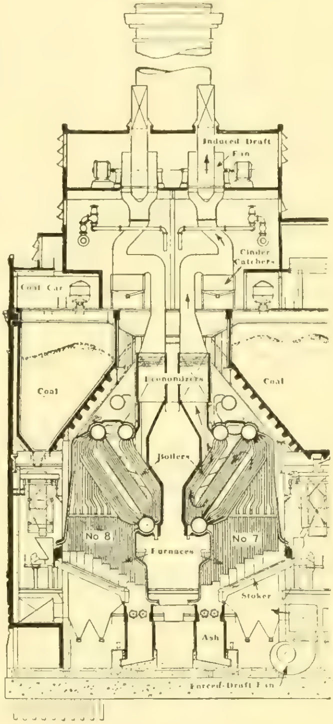

Fig. 546e.—Stoker-Fired Steam-Generating Units in the Hudson Avenue Station. (Units arranged back to back.)

(g) The illustrative tabular data and curves given in this chapter are more complete and have more items than are required when the units to be tested contain fewer elements, or are fired by hand, or when less extensive and detailed information will satisfy the requirements.

The discussion in this chapter has served to indicate in a general way how the efficiency of a steam generating unit is influenced by such factors as load, fuel, system of burning, and operating conditions, and to emphasize how dependent the net efficiency is on the energy consumption of the auxiliaries serving the unit. The various factors affecting the performance of steam generators will be considered in greater detail in the succeeding chapters.

CHAPTER XXXI

THE BURNING OF COAL ON GRATES AND STOKERS

398. Introduction.—(a) Consideration of a general character has already been given to fuels and their properties (Chap. XXVII), to the physics and chemistry of combustion (Chap. XXVIII), and to the relation between the combustion apparatus and the rest of the steam-generating unit (Chap. XXX). Attention will now be directed to the actual behavior of fuel while undergoing combustion within a furnace, to the apparatus used in the process, and to the conditions requisite for obtaining the most satisfactory results. Due to the widely differing characteristics of the fuels and to the great diversity of conditions under which combustion may take place, it will be impossible to give a detailed discussion of all the possible cases that might occur; however, there are certain general principles and typical forms of apparatus which, if understood, will be of great value in the solution of most problems arising in actual combustion engineering, and an attempt will be made to cover these in the discussion which follows.

(b) In Sect. **370,** which should be reviewed, it was shown that, regardless of the kind of fuel, there are **four main requirements** which must be fulfilled to ensure complete combustion: (1) ***air*** (oxygen) must be supplied in sufficient amount, (2) there must be a proper ***mixing*** of it with the fuel, (3) the ***temperature*** must be adequate to initiate and maintain the combustion process, and (4) there must be enough ***time*** allowed for the process to be completed, i.e., the *furnace* must provide a combustion space of sufficient volume and proper arrangement. These major requirements should always be kept in mind (as by means of the key word **matt,** p. 362) when considering combustion engineering problems. Either directly or indirectly, they will appear constantly in the following discussion.

399. How Coal Burns on Grates.—(a) In carrying out the combustion process with coal burned in bulk on an up-draft grate, or equivalent stoker, the air supplied for the process is generally divided into two parts, respectively known as the **primary air** and the **secondary air,** the first being passed *through* the fuel bed and the second being introduced *above* the bed. The actual reactions occurring during the process, which were given briefly in Sect. **374,** p. 372, are represented graphically in Fig. 547, in which the abscissas are the heights of various levels in the fuel bed and combustion space, measured in inches above the grate surface as datum. The curves are merely illustrative and show only in a general way how the gases may vary in composition and temperature as they pass through a 6-in. ash-free fuel bed and part of the combustion space. Referring to the figure, observe that the *primary air* ascending from the ash pit comes first into the **oxidizing zone** of the fuel bed, where practically *all* of its oxygen is consumed by reacting with the very hot carbonized coal in this region to form CO_2. This inert gas, together with the remaining nitrogen (not shown in the figure), then rises through the **reducing zone,** which extends to the surface of the fuel bed and in which a portion of the CO_2 is converted into CO by reacting with the carbon in this zone. The mixture of inert and combustible gases ascending through the upper part of the fuel bed, which constitutes the **distilling zone,** mingles with the volatile matter given off by the freshly fired coal in this region, and finally passes into the **combustion space** above the bed, where the *secondary air* is admitted. Here the combustion process is completed, provided there are adequate temperature, time, combustion space, and mixing of the gases and air. If the fuel bed were made very thick, the curves for the CO_2 and CO formed with the primary air would approach, as asymptotes, the lines for the equilibrium percentages that these conjugate constituents would have under the existing conditions; and the higher the temperature and the less the air velocity through the fuel bed, the farther apart will these asymptotes be. Although so shown in the figure, the zones are not, of course, sharply defined, but each gradually merges into the next, and the distillation zone is mostly in the reducing zone. As stated before, the figure is merely illustrative.

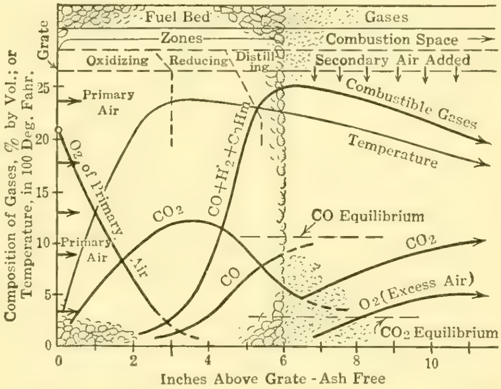

Fig. 547.—Composition and Temperature of Gases in the Different Combustion Zones When Coal is Burned on Grates.

(b) A series of tests, conducted by the U. S. Bureau of Mines on "Combustion in the Fuel Bed in Hand Fired Furnaces,"[1] with rates of combustion from 20 to 185 lb. per sq. ft. of grate surface and with fuel beds 6 and 12 in. thick, shows in general (although the other data varied considerably) that within the thinner bed most of the oxygen of the primary air was consumed in the first 3 to 4½ in. above the grate; that the percentage of CO_2 in the bed reached a maximum in about that height; that, as soon as formed, a little of the CO_2 is converted into CO, but that the more active conversion begins at a height of from 3 to 4 in. and continues to the surface of the bed where the equilibrium proportions are approached as closely as is permitted by the temperature and time of contact between the gases and the hot coke, both being dependent on the rate of combustion; that the maximum temperature is reached in the fuel bed at a height of from 3 to 5 in. above the grate; and that secondary air is needed above the bed to complete the combustion process, as no oxygen from the primary air will pass through the fuel. With the 12-in. bed the oxygen disappears 4½ to 6 in. above the grate. Regardless of thickness, the rate of feeding air to the fuel has practically no affect on the composition of the gases within the bed.

(c) Obviously, the **rate of burning** of the coal on the grate is controlled by the *primary* air, and the **completeness of combustion** is dependent on the *secondary* air; and if air passes into the combustion space through holes in the bed or by leakage around the grate, the amount of secondary air required from other sources is correspondingly reduced. Also, to complete the combustion as nearly as possible *within* the fuel bed itself and under conditions which cause a high temperature to exist, (1) the fuel bed must be relatively thin, (2) blowholes must not be permitted to form, and (3) the flow of air and gaseous products must be rapid, to reduce the time element. In addition, secondary air must usually be supplied *above* the bed. As a general statement, about half of the weight (approximately 15 lb.) of air used per pound of coal is supplied through the fuel bed and the other half above the bed.

400. The Amount of Excess Air Required.—**(a)** The weight of air just required for the complete combustion of a unit weight of any kind of fuel of known ultimate analysis can be obtained by means of Eq. (524) given on p. 371. With the usual **coals,** however, the **pounds of air required chemically** *per pound of this fuel* are given approximately by the empirical equation,[2]

$$w_a = 7.8(\text{Heating value, B.t.u. per lb. coal} \div 10{,}000) \quad . \quad (572)$$

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[1] Technical Paper No. 137, by Kreisinger, Ovitz and Augustine, 1916.

[2] See, "Limiting Factors in Reducing Excess Air in Boiler Furnaces," by E. G. Bailey, Mech. Eng., July, 1926, p. 706. Note that this equation does not apply to fuels other than coal.

The **actual weight of air needed** to ensure complete combustion throughout *all* parts of a body of fuel in a furnace is always greater than w_a as some *excess air* must invariably be supplied to avoid having a deficiency arise anywhere in the combustion region, either because of unequal distribution or due to imperfect mixing (loss from excess air being much less than that from an equal deficiency percentage, as was shown in Sect. **382** (j), p. 369). For example, when solid fuel is burned in mass on grates, the thickness, compactness and texture of the fuel bed are not uniform throughout the bed, thereby causing the air flow to be unequally apportioned; and even if the air present in any region would otherwise be adequate, it may not be properly mixed with the fuel but may be in separate stratifications or masses; hence, to ensure complete combustion of every part of the fuel in the short time available in a furnace, the weight of air supplied for the aggregate must be somewhat greater than that just sufficient for the carrying out of the chemical reactions when plenty of time is allowed for thorough mixing.

(**b**) The **percentage of excess air needed** depends on factors such as: (1) the composition and properties of the fuel, (2) the general method of burning the combustible, (3) the manner in which the air is supplied and mixed with the fuel in the different regions, (4) the arrangements and proportions used for the grate and combustion chamber, and (5) the furnace temperature desired, or that will be withstood by the furnace and related parts.

In general, the best percentage of excess air to use with a given coal and furnace must be found experimentally; its value is fixed by the original design and construction of the combustion equipment, and by the other factors just stated. In planning a new installation, after deciding on the kind of fuel and the combustion rate to be used, consideration must be given to the relations between the excess air percentage and the designs of the furnace and combustion equipment which it is possible to use, before selecting the combination which will most satisfactorily meet the conditions imposed.

When coal is burned on grates or with stokers, excess-air percentages of from 15 to 50 are the minimum values commonly used, but special conditions may require supplying even larger percentages. As will be seen later, pulverized coal, oil, and gas do not require so large a proportion of excess air as this.

(**c**) How the percentage of excess air, or the corresponding percentage of CO_2 in the exit gases, affects the **percentage of heat undeveloped** because of incomplete combustion of the CO, H_2 and hydrocarbons given off from the fuel bed, is illustrated by curve (*a*) in Fig. 548, which happens to be for Pocahontas Coal burned in a certain furnace with combustion conditions as given in the figure.[3] Clearly the loss from undeveloped heat is reduced by increasing the air supply.

But heat cannot be absorbed by the boiler of the usual type from the furnace gases below the temperature of the steam; hence, while an increase in the excess-air fraction causes more of the energy of the fuel to be developed, yet a greater proportion of this energy cannot be utilized for generating steam. In the figure, the line (*b*) gives the quantity of heat required to raise the temperature of the dry gaseous products of combustion to the saturation temperature of the steam. The total energy unavailable for the boiler due to both incomplete combustion and the high temperature of the dry exit gases is shown by curve (*c*), the lowest point of which determines the **optimum percentage of excess air,** or of CO_2, for this particular fuel, furnace and set of combustion conditions.[4] With other kinds of fuels, combustion rates, ratios of combustion space to grate area, etc., the curves and the optimum percentages may be quite different from those shown here, as will be seen later.

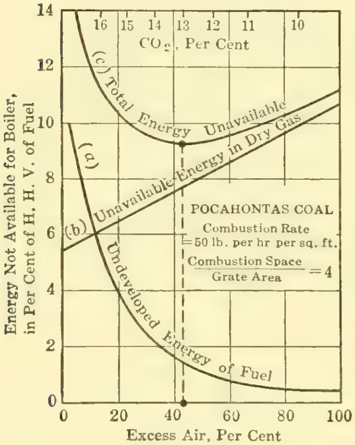

FIG. 548.—Energy Waste as Affected by Excess-air Percentage.

But the amount of air used also affects the *temperature* developed within the furnace, and in order to control this temperature properly an excess-air percentage different from the optimum value may be desirable.

401. The Furnace Temperature and Its Control.—(a) The furnace temperature has an influence on (1) the ignition of the fuel, (2) the rate of distillation of the volatile matter, (3) the proportion of CO formed, (4) the rate of combustion, and (5) the rates of heat absorption by various parts of the steam-generating unit. The control of the temperature within a furnace operating at any combustion rate is by means of the amount of excess air supplied.

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[3] Bureau of Mines Bul. 135, "Combustion of Coal and Design of Furnaces," by Kreisinger, Augustine and Ovitz.

[4] Variations in the moisture derived from the fuel and its combustion or from the air used, and differences in steam pressure, do not need to be considered in this connection as they do not change appreciably the shape or horizontal position of the curves, and hence do not affect the abscissa for the optimum excess air percentage.

(b) The **ignition temperature** of carbon monoxide and methane (CH_4) when mixed with *air* is from 1200 to 1400 deg. fahr., and for H_2 it is approximately 1100; and the *glow point* of bituminous coal occurs at about 850 deg. fahr., while for anthracite and coke it is above 1100 deg. fahr.[5] To ignite all parts of the fuel and to maintain combustion, the temperature of the fuel must be brought to and kept at or above these respective values at the proper points. The temperature to which the entering fuel is subjected can be increased by reducing the supply of excess air, by using heated air, or by providing in the furnace suitable wall surfaces or arches which radiate back to the fuel much of the energy they receive from hotter portions of the fire, by exposing to the fuel less of the relatively cold surfaces of the boiler and water walls, or by using a combination of these methods. To ignite and burn fuels of low volatility and also the volatile matter rising from the distillation zone, the heat returned by radiation should be concentrated at the places in the furnace where the fresh fuel is being fired. With the higher rates of combustion, the furnace temperature is somewhat greater than with more moderate ones; but with a given furnace there is, of course, a temperature which should not be exceeded.

(c) A **limitation of the furnace temperature** may be necessary because, if the temperature is too high, (1) furnace walls made of refractory material will soften and deteriorate—especially if they are tall, of unsuitable grades of fire brick and not adequately cooled; (2) the ash in the fuel bed may fuse into clinkers and may adhere to walls and grates or stoker parts, thus causing operating difficulties and increased expenditure for maintenance; and (3) molten fly-ash, carried in suspension by the furnace gases, may be deposited on refractory walls and cause their erosion by flowing over their surfaces, or this ash may adhere to improperly spaced boiler tubes and bridge between them, thus interfering with the draft and lowering the steaming capacity of the boiler unit.[6] By using a sufficient amount of excess air the furnace temperature can be lowered to a safe value, and higher rates of combustion are made possible, but with some sacrifice in efficiency. When preheated air is supplied it may be especially necessary to use rather a large dilution factor, although the gains from preheating are somewhat reduced thereby.

[5] The definition of just what is meant by ignition temperature, and the methods of measuring this quantity, are yet to be standardized; hence the values given are merely illustrative.

[6] How these difficulties can be minimized, and greater combustion rates be made permissible by proper design and arrangement of the apparatus, will be considered later when the structural features of furnaces and boilers are discussed. (Chaps. XXXVII and XXXVIII.)

(d) The **influence of the furnace temperature on the rates of heat absorption** of the various parts of the unit may be shown as follows: If the temperature within a furnace evolving heat at a given rate is increased by reducing the air dilution, or if the temperature is raised due to using a higher combustion rate, then (1) a larger proportion of this heat will be absorbed as radiant energy by the boiler tubes and furnace water-walls (if any) exposed directly to the fire, and consequently (2) a smaller proportion of the heat will be carried by convection to the boiler tubes and other heat absorbing surfaces located farther along in the gas path; thus (3) if a *convection* type of superheater is used, the steam will be superheated to a lower temperature, whereas, with a superheater of the radiant type (which is placed directly in the furnace walls) the steam temperature is increased, and (4) economizer and air preheater elements (if used) would receive less heat, hence the temperatures of the feedwater and combustion air would be reduced. Thus, the functioning of the related apparatus may make it necessary to control the furnace temperature by varying the excess air percentage with respect to the combustion rate, and to use a value different from the optimum percentage found by the method given in Sect. **400 (c)**.

402. The Combustion Space Required.—(**a**) The space needed within a furnace to provide the *time element* necessary for properly burning the volatile and gaseous fuel issuing from the fuel bed depends, as to volume and proportions, mainly on (1) the amount and character of the volatile matter in the fuel burned, (2) the percentage of excess air used, and (3) the rate of combustion—all of which affect the volume and length of the flame issuing from the bed. The size and arrangement of furnace space should be such (*a*) that the flame will not be chilled and extinguished by coming into contact with relatively cold objects, such as the boiler tubes, before combustion is completed; and (*b*) that all of the space will be used as fully as possible. The length of flame for which provision should be made varies from a few inches with coke or anthracite burned at low rates on grates or stokers to 40 ft. or more with volatile coals consumed at high rates with such apparatus.

(**b**) Less combustion space is required when special provision is made, as is done with some stokers, for having the air properly apportioned to the different parts of a fuel bed that may vary in thickness, and when there is special means of mixing the secondary air and the gases— such as by injecting the secondary air in numerous small streams at high velocity, employing steam jets above the fire, or using baffle walls, arches or piers in the flame passage.

403. Influence of the Physical Characteristics of the Fuel.—The *physical characteristics of the fuel*, in addition to the chemical properties, may also have an important influence on the equipment and operating conditions needed for properly carrying out the combustion process. Some coals are free burning and can be consumed satisfactorily while in a quiescent state, as shown in Sect. **352**, whereas other coals "cake" when heated and impede the passage of air through the fuel bed unless subjected to agitation which constantly breaks up the fused mass as rapidly as it forms. Some coals have an ash content which fuses only at high temperatures and therefore is easily disposed of; while the ash of other coals softens at low temperatures and may consequently cause operating difficulties and carry a considerable amount of entrapped carbon with it to the ash pit, especially when intensive combustion is used. Some coals, e.g., anthracite, must be of uniform size to burn with satisfactory results, and other coals burn best as run-of-mine or as slack. Thus, often the kind of stoker, or the method of hand-firing on grates, must be selected with respect to the physical behavior of the coal and its ash while undergoing the combustion process.

404. "Pittsburgh Tests" by the Bureau of Mines.—(**a**) Because the combustion space provided has such an important influence on the combustion process, the U. S. Bureau of Mines constructed a special experimental furnace with which it conducted about one hundred tests to obtain, among other things, quantitative information as to the furnace volumes and lengths needed when burning three representative coals under various combustion conditions.[7] As the results of these tests are very enlightening and constitute at present almost the only available information at all comprehensive on the subject, they will be reviewed briefly. These tests were made with an *overfeed stoker*; hence they do not apply directly to other types.

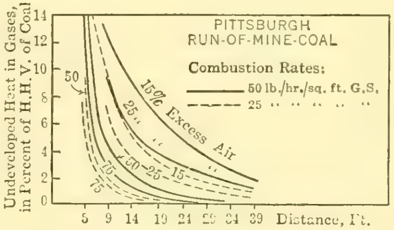

FIG. 549.—Completeness of Combustion along the Path of the Gases under the Conditions Given in the Figure.

(**b**) The tests showed that the *percentage of undeveloped heat* of the combustible gases rising from fuel beds was affected by the distance traversed by the flame, by the percentage of excess air, and by the rate of combustion, in a manner illustrated in a general way by the curves in Fig. 549, which is for Pittsburgh run-of-mine coal. This figure clearly emphasizes the importance of having proper size and proportions for the combustion space, and also shows that (1) the heat release occurs rapidly in the first part of the combustion space, but that (2) the rate quickly decreases as the gases move away from the fuel bed. To burn the last 1 or 2 per cent of the combustible gases would require such great increase in furnace volume as not to be practicable; thus there is a certain percentage of undeveloped heat justified, the amount in each case depending on the local conditions and economic factors involved.

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[7] Bureau of Mines Bul. 135 referred to in footnote 3, p. 462. The apparatus used consisted principally of a Murphy stoker (somewhat like that in Fig. 566) having 25 sq. ft. projected grate area, and a horizontal, tunnel-like combustion space 3 × 3 ft. in cross-section and 40 ft. in length, with observation and sampling holes located along it at 5-ft. intervals. Gas samples were also taken at points close to the fuel bed. The compositions of the coals used are given in Table XLVIII.

TABLE XLVIII

Chemical Characteristics of the Three Coals Tested

Item No.	Item	Pocahontas Coal	Pittsburgh Coal	Illinois Coal
1	Volatile matter in moisture-and-ash-free coal, per cent	18.05	34.77	46.52
2	Fixed carbon in moisture-and-ash-free coal, per cent	81.95	65.23	53.48
3	Carbon in moisture-and-ash-free coal, per cent	90.50	85.7	79.7
4	Volatile carbon in moisture-and-ash-free coal, per cent	8.55	20.47	26.22
5	Available hydrogen in moisture-and-ash-free coal, per cent	3.96	4.70	3.96
6	Ratio of volatile carbon to available hydrogen, per cent	2.16	4.35	6.6
7	Oxygen in moisture-and-ash-free coal, per cent	3.32	5.59	10.93
8	Nitrogen in moisture-and-ash-free coal, per cent	1.19	1.73	1.70
9	Percentage of moisture accompanying 100 per cent of moisture-and-ash-free coal	2.53	2.88	22.07
10	Volatile matter times the ratio of volatile carbon to available hydrogen (product of items 1 and 6), per cent	39	151	307
11	Ratio of oxygen to total carbon, in moisture-and-ash-free coal	0.0367	0.0652	0.137
12	Total moisture in furnace per pound of coal reduced to moisture-and-ash-free basis, pounds	0.409	0.501	0.700

(c) The relations, found to exist between the relative size and length of combustion space and the *completeness of combustion* for the three fuels tested, when burned at different rates and with various percentages of excess air, are indicated in the abbreviated chart given in Fig. 550. To illustrate the use of the chart: Suppose a furnace is to be designed to burn coal at the rate of 40 lb. per hr. per sq. ft. of grate surface (G. S.), with 50 per cent excess air and with an incompleteness of combustion of only 2 per cent of the H.H.V. of the coal as fired. Then the chart shows that the Pocahontas, Pittsburgh and Illinois run-of-mine coals would require respectively 3.2, 3.9 and 5.8 cu. ft. of combustion volume per sq. ft. of grate surface, and that the corresponding furnaces should provide paths nearly 10, 12 and 18 ft. long for burning the gases under these conditions.

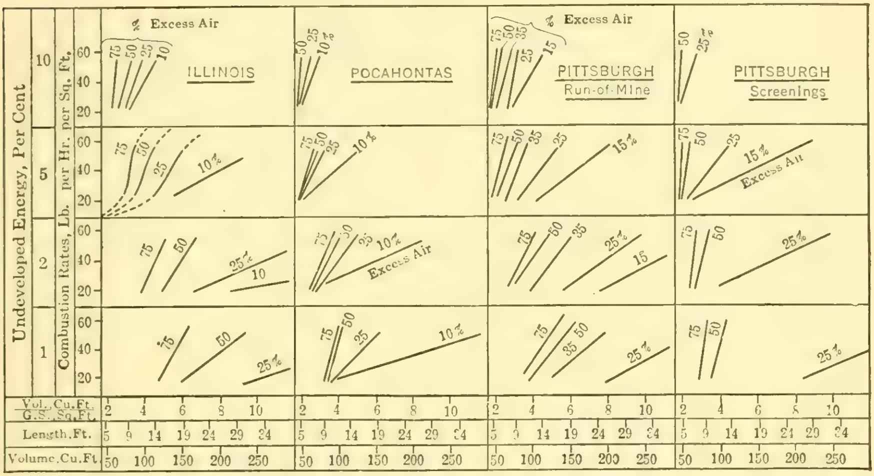

FIG. 550.—Relation between the Required Size of Combustion Space, Given Completeness of Combustion, the Rate of Firing, and the Excess-air Percentage. (Pocahontas, Pittsburgh and Illinois Coals.)

(d) The *optimum percentage of excess air* was found to be influenced by the furnace volume and by the kind of coal burned, in the manner shown in Fig. 551 for a combustion rate of 50 lb. of coal per hr. per sq. ft. of grate surface. This chart resembles Fig. 548 in its construction; and on it the figure (3), (5) or (7) placed near a curve gives the number of cubic feet of combustion space per square foot of grate surface for which the curve applies. Observe that, with a given coal, the *larger the size of the furnace* (1) the less is the amount of heat wasted and (2) the smaller is the optimum percentage of excess air. The effect of the compositions of the different fuels is indicated by the relative positions of the corresponding curves.

Figure 552 gives the combinations of furnace size and excess air percentages which were found in the tests to give the minimum losses with the different coals, each burned at the rate shown.

(e) With the arrangement of stoker and combustion space used in the tests, the gases and secondary air were rather thoroughly mixed and were kept at high temperature until combustion was completed. With other fuels, types of grate or stoker, furnace arrangements, and operating conditions, the data given in Figs. 549 to 552 inclusive may not apply directly; but if none better are at hand (and unfortunately the data now available are meager and limited in scope), new designs may be based on these curves after introducing modification to make them accord with the expected differences in conditions.

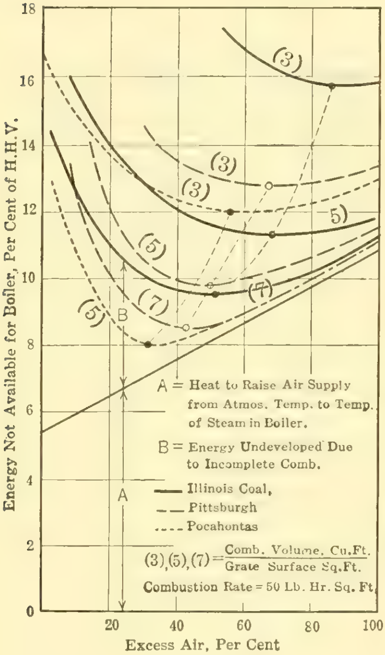

FIG. 551.

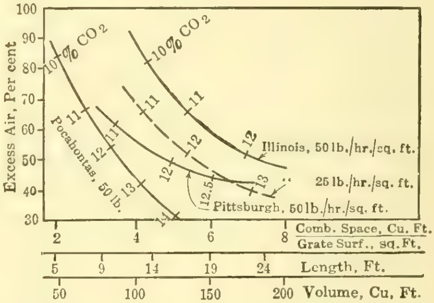

FIG. 552.—Furnace Size and Percentage of Excess Air Giving Minimum Heat Losses.

When **making modifications** the following observations, based on the analysis of the test data, may be of value: (1) The size of the combustion space needed does not vary directly as the *quantity* of the volatile matter distilled off by a given coal when burned with different rates of combustion; for when the rate of burning (and of distillation) is doubled, the furnace volume need be increased only between 20 and 40 per cent to maintain the same completeness of combustion with the same excess-air percentage. (2) With different coals the combustion spaces required are not in the same proportions to each other as are the *percentages of volatile matter* in the coals; for with the Pocahontas, Pittsburgh and Illinois coals tested, the volatile contents are to each other as 1:1.92:2.56, whereas combustion spaces in the proportions 1:1.2:1.8 are needed if 98 per cent completeness of combustion is desired in each case when using 50 per cent of excess air and combustion rates of from 25 to 50 lb. per hr. per sq. ft. of grate surface. (3) The *quality* of the volatile matter also has its influence; for, in general, the higher the carbon content in the volatile hydrocarbon compounds the longer is the time (and the longer is the space) required for completing their combustion. (4) For nearly complete combustion with 50 per cent excess air and a 50-lb. combustion rate, the volume of combustion space needed is about 3 + [per cent volatile combustible × (volatile carbon ÷ available hydrogen)] ÷ 100, cu. ft. per sq. ft. of grate surface.[8]

405. General Requirements of Combustion Equipment and Its Operation.—(**a**) In **selecting combustion equipment,** or in designing it, provision should be made to have under the *normal* conditions of operation as efficient generation of heat as the *cost factors* warrant, and at the same time have sufficient capacity for carrying any expected *maximum demand,* and also ability to meet the sudden load changes as rapidly as they may occur. As already seen, the important factors to consider are: (1) the grate area, and (2) the draft for supplying the primary air necessary to produce the proper rates of combustion; (3) provision for the supply of secondary air; (4) means for regulating and properly apportioning the primary and secondary air; (5) arrangement for mixing the secondary air and gases; (6) means for maintaining the combustion of the gases; (7) a combustion space of such volume and proportions as will provide for the length of flame (i.e., give sufficient *time* for the combustion of the gases to be completed); (8) means for overcoming difficulties arising from any caking of the fuel or clinkering of the ash that may occur; (9) the various items entering into the first cost of the equipment; (10) the cost for labor, repairs and maintenance; and (11) the general suitability and convenience.

(**b**) In the **operation of a furnace** the maximum efficiency is most easily obtained when the combustion conditions are maintained as nearly unvarying as possible For best results (*a*) the fuel should be uniform in character; (*b*) the rate of feeding should be constant; (*c*) the evolution of the gases should be unchanging in character and rapidity; (*d*) the primary and secondary air should be correctly apportioned and kept at the proper fixed ratios with respect to the fuel burned and to each other; (*e*) the thickness and character of the fuel bed should be maintained uniform; (*f*) the caked masses formed (if any) should be continually broken up; and (*g*) the ash and clinker should be disposed of at a uniform rate.

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[8] For more complete and more accurate data and a discussion of other factors involved, consult the original source, Bulletin 135, or Haslam and Russell's "Fuels and Combustion."

How closely these desired conditions can be approached in hand-fired and stoker-fired furnaces will be observed in the discussion which follows.

406. The Burning of Coal on Hand-Fired Grates.—(a) The more important factors to be observed in the operation of furnaces were given in the preceding paragraph, and emphasis was placed there on the desirability of maintaining the combustion conditions *uniform.* In hand stoking, however, the various operations are performed *intermittently*, and thus the conditions are constantly varying. Obviously the best condition of the fuel bed is obtained (1) when the coal is fired frequently, in small amounts, and with proper distribution; (2) when the caked masses of coal (if any) are broken up as rapidly as formed; and (3) when the ashes and clinker are not allowed to clog the bed.

(b) In general, there are three **methods of hand firing** commonly used:

(1) In one, called *spread firing*, a small amount of the fresh coal is each time distributed evenly over the entire surface of the bed. This is the method commonly adopted with anthracite and other low-volatile coals.

(2) In the second, known as *alternate firing*, the new coal is placed on but part of the grate at a time so that the freshly evolved volatile matter will be exposed to the other brighter parts of the fuel-bed surface which provide the necessary heat for supporting the combustion of these gases. The coal may be placed alternately on one half of the grate at a time, or on alternate strips, or on alternate spots of an imaginary checker board—each time being distributed over the brighter and thinner portions of the fuel bed—thereby also keeping the bed uniform in thickness and avoiding the formation of holes in the fire. This method of stoking is particularly suitable for non-caking coals. However, caking coal can also be burned satisfactorily with this method provided the fuel is fired frequently and in rather thin layers; for under these conditions the crust formed will burn through and disintegrate quickly thus offering but little impedance to the air flow through the bed.

(3) In the third method, known as *coking-firing*, which is especially suitable for use with caking coals, the fresh fuel is placed on the front edge of the fuel bed and allowed to coke there, the volatile matter passing back over the hot bed. After the distillation is completed the carbonized fuel remaining is pushed back and distributed over the bed. This

method, while effective, does not permit obtaining as high rates of combustion as with the other methods.

The conditions within a hand-fired furnace are illustrated in Fig. 553, the thin spot at (*a*) being the place where fresh coal should be added to equalize the thickness of the bed. The figure also shows a typical arrangement of the furnace and its fittings.

(c) In **apportioning the air** used for combustion, the ash-pit-door opening and the grating in the fire door should be so adjusted that the air supply is properly divided into its primary and secondary portions. Usually about half of the air should pass through the fuel bed and half be admitted over the fire, the total amount per pound of coal being about 15 lb., or 200 cu. ft. under normal atmospheric conditions. The best method of regulating the draft to meet the varying rates of combustion due to load fluctuations is by means of the damper in the outlet of each boiler, leaving undisturbed the amounts of opening provided by the ash-pit and fire doors. The control may be by hand or by means of automatic damper regulators, which are operated by the variations in the steam pressure accompanying unequal changes in the load and rate of steam generation. With a *hot* fire the volatile matter from a thin layer of bituminous coal is distilled in from two to five minutes after firing, and during this period a larger amount of secondary air is needed than should be supplied after distillation is completed. The rapid variation in the requirements makes the proper regulation of the secondary air difficult or impossible. The best condition can be obtained when the fresh coal is always added at or shortly after the completion of the distillation of the previous charge, the best *intervals between firing* being

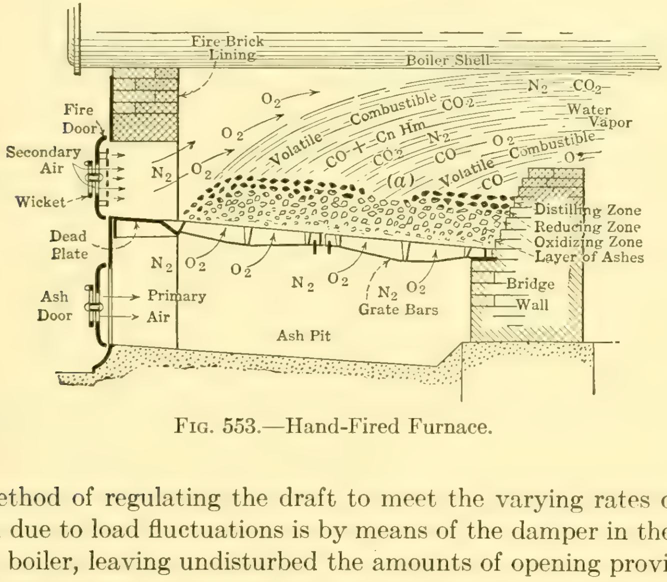

Fig. 553.—Hand-Fired Furnace.

from three to eight minutes with bituminous coals. With the less volatile anthracite and semi-bituminous coals the firing can be done less frequently. While the gases are being evolved most rapidly the fire door may be left partly open to supply the extra secondary air needed, or automatic devices, which remain open immediately after firing and then gradually close the necessary amount, may be used for the same purpose. The time during which the fire doors are open for adding fuel, breaking up caked coal or cleaning the fire should be reduced as much as possible so as to minimize the excessive amount of air that enters during such periods.

(d) **Cleaning the fire** becomes necessary at proper intervals because of the constant accumulation of the clinker and coarse ash that are unable to pass through the openings in the grates; there are two methods in common use. In the *side method* the good coal is first scraped and pushed to one side, and then the clinkers on the bared half are loosened from the grates by means of a slice bar and are scraped out of the furnace by using a special hoe. After one side is cleaned, all the burning coal is transferred to it and a little fresh coal is added; then the remainder of the grate is cleaned. Finally the burning coal is uniformly redistributed and normal conditions are re-established. In the *front-to-rear method* of cleaning, the burning coal is pushed back against the bridge wall by means of the hoe, and the grate is then cleaned, preferably treating one half at a time. While one cleans the fire, the damper should be partly closed to avoid cooling the boiler by the large inrush of cold air.

To attempt to clean a bed of clinkering coal without first displacing the burning mass is faulty practice; for with such a method only the larger clinkers can be removed, and in using the slice bar and rake the ash and smaller pieces of clinker remaining are brought into the high-temperature zone of the bed where they fuse, form additional clinkers, perhaps worse than those removed, and may even become sufficiently molten to flow into the grate openings.

(e) **Clinker trouble** is aggravated by using fires unnecessarily thick, because they reduce the amount of air flowing through the grates and require that a larger proportion of the total air supply be delivered above the bed to burn the increased quantity of CO issuing therefrom under these conditions. The lower part of the fuel bed, and the ashes therein and immediately below, are therefore at a temperature higher than if a thinner fire were used. Clinkering is also promoted by the presence of burning coal in the ash pit and by the use of preheated air. If the ash is the kind that fuses at low temperature, these conditions that contribute to clinkering should of course be avoided, and every effort should be made to have the ash on the grates and the lower part of the fuel bed maintained relatively cool. Sometimes it is advantageous to keep water in the ash pit to quench the ash; and, though otherwise inefficient, it may even be advisable to blow steam under the grates, preferably using for the purpose steam that has been used in pumps or other apparatus and which would otherwise be wasted.

The absorption of heat in superheating the steam and in partly decomposing it into oxygen and hydrogen probably aids in keeping the temperature of the ash below the melting point.

(f) The **thickness of fuel bed** that will give the best results depends on many factors, including the furnace design, the kind, size and condition of the coal, the characteristics of the ash, the draft available and the loading. This thickness is best determined in each case by experiment, especially when analogous data are not available. In general, with natural draft, thicknesses of from 4 to 8 in. are commonly used with run-of-mine bituminous coals and with anthracite buckwheat; while with semi-bituminous coals they may be from 10 to 14 in. With heavy loading it may be desirable to use a relatively thin fuel bed so as to increase the flow of primary air obtainable with the draft available.

But no matter how carefully a furnace may be operated, efficient and smokeless combustion cannot be obtained unless combustion space and grates of suitable size and arrangement are provided.[8a]

407. Combustion Chambers Used with Hand-Fired Grates.—(a) As the portion of the furnace within which the gases from the fuel bed are burned has an important influence on the efficiency of combustion and on the production of smoke, its design should be given careful consideration. In general, in the hand-fired units this chamber is provided with fire-brick walls which become heated and aid in maintaining the high temperature required over and beyond the fuel bed; and often the combustion chamber is given special form for promoting the mixing of the gases with the secondary air. Many different designs are in use or have been proposed to meet the widely different conditions encountered; and only brief discussion of a few typical arrangements can be undertaken here.

(b) With anthracite coal the combustion space may be relatively small and simple in arrangement, although sometimes a fire-brick arch sprung over the grate is desirable to assist in the ignition of the freshly fired fuel. Fig. 554 shows a simple water-tube boiler-setting with *roofless* combustion space suitable for anthracite and other coals rather low in volatile content.

[8a] Anyone responsible for the operation of hand-fired furnaces should consult the following: Bureau of Mines publications, especially Technical Paper No. 80, "Hand Firing Soft Coal Under Power-plant Boilers," by Henry Kreisinger; Univ. of Illinois Engineering Experiment Station Circular No. 7, "Fuel Economy in the Operation of Hand-Fired Plants"; and The Cochrane Handbook on "Finding and Stopping Waste in Modern Boiler Rooms," The Cochrane Corp., Philadelphia.

(c) With the more volatile coals the length of passage through the combustion space should preferably be made as long as the flame. A **Dutch oven,** as shown in Fig. 555, may be placed in front of the boiler to give long passage for the burning gases and to provide a large volume and an arched fire-brick *roof* that reflects and radiates heat to the fuel bed and gases arising therefrom. While intended to provide for the smokeless combustion of high-volatile coals, experience shows that when large amounts of such coal are fired, the intense heat from the arch causes the gases to distill so rapidly that puffs of dense smoke issue despite all efforts to prevent their occurrence. Dutch-oven arrangements are, however, suitable for burning coals and other fuels (such as wood, bagasse, etc.), of high moisture content; and their extra space requirements can be reduced or completely eliminated by placing them under the boiler, although the effectiveness of some of the heating surface is sacrificed thereby.

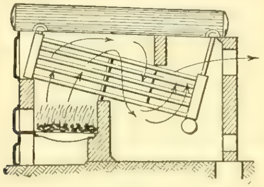

Fig. 554.

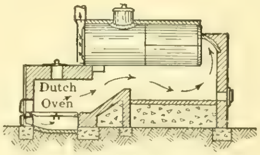

Fig. 555.—Dutch-Oven Furnace

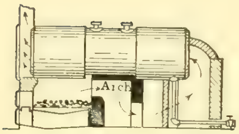

Fig. 556.

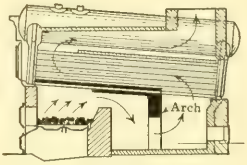

Fig. 557.

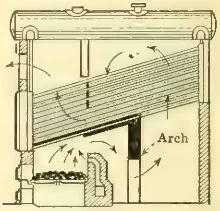

Fig. 558.

Figure 556 shows a modification of the Dutch oven in which the arch is moved back so as to expose to the fuel bed some of the surface of the boiler (which here is of the fire-tube type), a deflecting arch having also been added to aid in the mixing. Figures 557 and 558 illustrate somewhat similar arrangements with water-tube boilers having respectively horizontal and vertical baffles for the gases. In other arrangements the mixing is accomplished by means of multiple arches, piers, or wing walls; by injecting jets of air or steam through the front, sides, or bridge wall of the furnace; or by using various combinations of such expedients.

(d) Fig. 559 shows a Hawley **down-draft setting** having above the ordinary grate a second one composed of tubes which are kept cool by having water from the boiler circulate through them. The fresh coal is fired onto the water grate and when partly consumed falls to the lower one. The main supply of combustion air enters through the upper fire-door, travels downward through the green coal and joins the volatile matter distilling therefrom; the gaseous mass resulting then descends through the burning fuel on the bottom of the upper bed, thereby becoming highly heated and ultimately mixed, and finally passes through the combustion space formed between the two fuel beds, where it is exposed to the incandescent coals below. If the rest of the combustion chamber is properly designed, this arrangement of furnace will give efficient and smokeless combustion when properly operated, but its forcing capacity is limited. The down-draft arrangement is best suited for burning lump coal at low combustion rates, and requires the use of a strong draft.

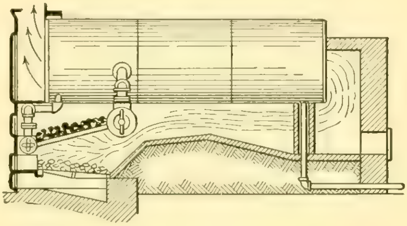

FIG. 559.—Down-Draft Setting.

408. Combustion Rates and Grate Areas for Hand Firing.—(a) The rate at which coal is burned in a furnace depends on such factors as the characteristics of the coal burned and those of the ash, the thickness and condition of the fuel bed, the total surface and air-passage area provided by the grates, and the draft. The most satisfactory combustion rate to use with a given coal and furnace is best determined experimentally. For example, Fig. 560 illustrates in a general way how the value of one kind and size of coal varies when burned in a given furnace at different rates, the thickness of fuel bed being maintained fixed. From a study of such curves for various values of draft

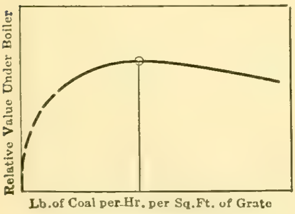

FIG. 560.

and bed thickness, the optimum rate of combustion per square foot of grate surface per hour, and the other operating conditions giving the best results, can be determined for that furnace. But unless the proper grate area has been provided in the original equipment of the unit, the required boiler output will not be met under these most favorable conditions. Obviously, a new installation should be provided with the size of grate most suitable for meeting the demands with any one of the various kinds of coal that it may eventually be desirable or necessary to use; and generally, the area should be sufficiently extensive to permit carrying the maximum load with the poorest coal that may have to be used under the most unfavorable emergency conditions that may be encountered. The selection of the size of grate is of course based directly or indirectly on the total fuel consumption expected per hour and on an allowed combustion rate per square foot of grate surface (G. S.) per hour, using for the latter such value as experience or available data may indicate to be the best for the probable conditions to be encountered and for the kind of coal to be used.

(**b**) As the volatile matter evolved from the coal is consumed mostly *beyond* the fuel bed, the rate at which coal can be burned *on* a given grate area is largely dependent on the proportion of fixed carbon in the fuel. With natural draft, from 12 to 16 lb. of *fixed* carbon are generally burned per square foot of grate surface per hour under the usual conditions, and grate areas determined on this basis ordinarily allow for overload capacities of from 50 to 100 per cent with the intensity of draft available with the usual chimneys. With forced draft, higher combustion rates can of course be used. With anthracite coal there must be a larger grate area for a given total rate of energy liberation than is required with bituminous coal.

(**c**) With hand firing, the **average rates of combustion,** in pounds of coal per square foot of G. S. per hour, used with natural draft and with forced draft (the latter given in parentheses) are about as follows: Anthracite, 15(20); Semi-anthracite, 16(25); Semi-bituminous, 18(35); Eastern bituminous, 20(30); Western bituminous, 30(35); Lignite, 25(35). Coke breeze is suitable to burn only with forced draft, the average combustion rate being about 20. In locomotives and marine boilers rates as high as 225 have been obtained; but, while they indicate what is possible, such rates cannot be considered for efficient operation. Some indication of the relation between the draft through the fuel bed and the combustion rates with various coals is given by Fig. 561.

(**d**) Formerly, when the heat-absorbing surface in a unit was only that of the boiler, and when the operating conditions were moderate, the grate area was quite commonly made a certain fraction of the boiler heating surface (e.g., from $\frac{1}{35}$ to $\frac{1}{60}$, depending on the kind of coal and other factors), but under modern conditions this method may lead to unsatisfactory results and generally should not now be used.

(e) The total width of grate surface is usually made equal to the distance between the walls of the boiler setting, and the length is made sufficient to give the desired area. With hand firing, *grate lengths* up to 10 or 12 ft. have been used with anthracite coal with dumping grates. Soft-coal fires are more difficult to handle, and with ordinary grates the limit is usually 6 ft.; otherwise there is difficulty in cleaning the fire.

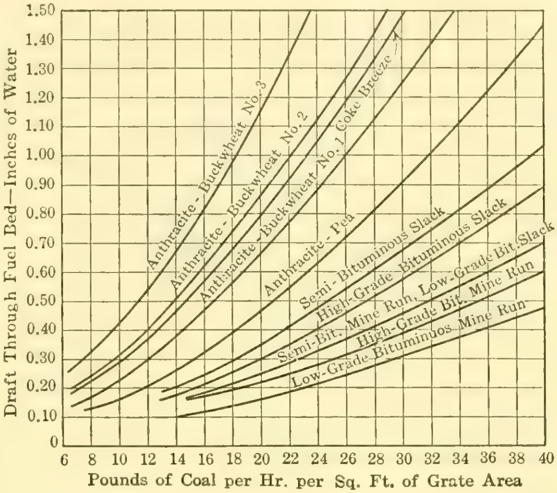

FIG. 561.—Relation between Draft and Combustion Rates of Various Coals Burned on Grates.

409. Grates.—(a) Grate bars, while primarily used to support the fuel bed, must also provide suitable passages for the admission of the primary air and for the free voidance of the ashes. To prevent the bars from burning away they should be made of suitable material (special C. I. is commonly used) and should have such shape as to present relatively small areas to the fuel bed and expose large surfaces to be cooled by the currents of primary air. Usually they have deep ribs not only for strength but also to provide the necessary cooling surface. The bars should be in short lengths (ordinarily not over 3 ft.) and so set, Fig. 553, as to allow for expansion, twisting, warping and sagging due to their wide variation in temperature and possible growth in size under prolonged heating. The width of the air passages through the grate should be made as large as the character of the coal and its ash permit without allowing unburned coal to drop through.

(b) Fig. 562 shows some typical forms of *stationary grate bars* for coal, and one (*c*) for sawdust. The flat plate grate at (*d*), which is for fine coal, has many small perforations between which the exposed surface is recessed so as to become filled with a permanent layer of fine ashes that will protect the bar from the heat and also prevent the adherence of clinker to the metal. Hollow grate bars are sometimes used in combination with a blower system, especially for burning sawdust, chips, shavings, tanbark or bagasse.

(c) Fig. 563 shows one arrangement of *shaking grate*, of which there are many different forms in use. With such grates the fire is more easily cleaned than with stationary grates and the furnace doors do not have to be opened during the operation. With some, the fuel bed is disturbed but little during the shaking process, which is a desirable feature when anthracite and some non-caking coals are burned; while others have means for agitating the bed, for breaking the clinker, and for dumping. They are generally arranged in sections which can be operated independently or together. Their cost is, of course, greater than for ordinary grates but they often give from 1 to 5 per cent better efficiency.

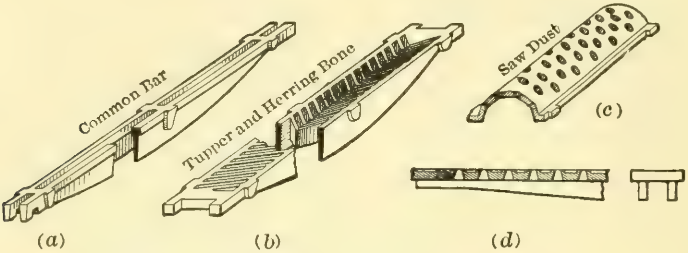

Fig. 562.—Some Typical Forms of Grate Bars.

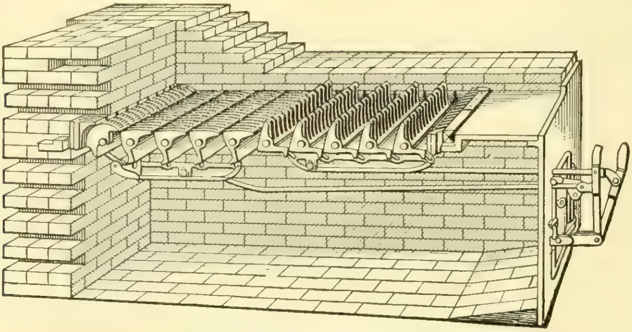

Fig. 563.—Shaking Grates.

(d) Inclined shaking grates can be arranged to feed the coal progressively from the front to a rear dump, and such arrangements are designated as *hand-stokers,* of which there are several types on the market. They simulate the action of the mechanical stoker, which is considered in the next section, and give better combustion efficiency and less smoke than is obtainable with hand firing; but, because of their relatively high initial cost, their use is not always economical, especially with boilers of the smaller sizes.

(**e**) But even with the best grates, *hand firing* is arduous and disagreeable work and requires close attention when properly done, especially when the loading is heavy; also there are inherent limitations to the combustion capacity obtainable with hand firing in a single steam-generating unit, and to the amount of coal that one attendant can stoke per hour, the man's firing capacity being dependent on the amount of effort he also gives to other duties such as maintaining the fuel bed in good condition, hauling the coal to the boiler, wheeling away the ashes, attending to the pumps and performing other secondary duties. Thus, with hand firing, the size of the individual steam-generating unit is restricted, and for a large total capacity many small units and numerous attendants are required. The other methods of firing yet to be considered permit obtaining in a single unit a much greater combustion capacity, with equal or better efficiencies, less smoke and less labor; also a greater amount of steam can be derived from a given size of boiler; or, for the same output, a smaller boiler may be installed, thereby reducing the cost of this part of the unit. But, while these other methods have certain advantages, nevertheless the use of hand firing is often justified, especially in small units.

410. Mechanical Stokers—General.—(**a**) Mechanical stokers are devices that permit solid fuels to be burned more efficiently, more rapidly, and with less labor than by hand firing; they are designed to receive fuel continuously by gravity, constantly feed it into the furnace, carry it progressively through the distillation, oxydizing and reducing zones of the fuel bed, and either discharge the ash automatically to the ash pit or deposit it where it can be readily disposed of with the minimum of effort. Equipped with suitable adjustments for regulating the rate of fuel feeding, thickness of bed and air supply, and supplemented by properly designed combustion space, they afford highly efficient means for burning coal in mass, provided difficulty does not arise from caking of the fuel or from clinkers. Some stokers are well suited for burning caking coals, some for handling clinkering ash, while others are applicable only when free burning and non-clinkering coal is used. Practically any kind of coal can be burned efficiently and smokelessly with some type of stoker; but no single design of stoker is equally suited for all kinds of coal, although some stokers will handle satisfactorily quite a wide range. As the coal is usually delivered to the stoker by some form of mechanical conveyor and as the ashes are also disposed of by similar means, little or no arduous physical labor is connected with the normal operation of stoker-fired units; but intelligent attendants are required in order to give proper attention to the regulation and care of the apparatus.

(**b**) The **principal advantages** derived from the use of mechanical stokers, as compared with intermittent hand firing, are: (1) the continuous delivery of the coal under uniform conditions; (2) the progressive and gradual distillation of the volatile matter, which facilitates making proper provision for the secondary combustion; (3) the ability to obtain better performance and smokeless combustion,—because the conditions make it easy to regulate carefully the operations at all times; (4) the greater combustion capacity obtainable in a furnace; (5) the flexibility of operation or ability to meet varying demands quickly; (6) the ability to burn poorer and cheaper grades of coal with less smoke and higher efficiency; (7) the relief of attendants from strenuous duties and from exposure to radiation from the furnace, thus eliminating the personal element in actual firing and permitting more careful attention to adjusting the operating conditions; and (8) the decrease in labor costs in large plants, through the reduction in the number of attendants needed. In small plants, however, there may be no reduction in the size of the firing force because a certain number of attendants are necessary anyway, and the introduction of mechanical stokers may not reduce this number.

(**c**) Commercial stokers are classified broadly as either (A) **overfeed stokers** or (B) **underfeed stokers,** depending on whether the coal is fed to the fuel bed above or below the level at which the primary air is admitted. In the first of these the burning fuel is in the lower part of the bed to the top of which the fresh coal is added; whereas, in the second, the incandescent fuel is on the top of the bed and the green coal is fed from below. The *overfeed type* includes (*a*) the **front-feed inclined-grate stokers,** which receive fuel at the front and feed it down an incline to the ash dump at the bottom, (*b*) the **double inclined side-feed stokers,** which feed from both sides onto grates that are inclined downward and meet at the ash pocket at the center, and (*c*) **chain-grates** or **traveling-grates** which continuously convey the coal bed horizontally from front to rear. The *underfeed type* includes (*d*) **single-retort stokers,** which are usually horizontal and may have lateral ash dumps, and (*e*) **multiple-retort stokers** which are usually inclined and have ash discharge at the rear. Some stokers are designed to operate only with **natural draft** and others require the use of **forced draft.**

A brief discussion of some of the more common forms of stokers and their application will now be given.

411. Overfeed Stokers.—(**a**) In most inclined overfeed stokers (see Fig. 564) the coal is deposited in a hopper from which it is mechanically and continuously fed to the grate and made to pass under a more or less extensive *coking arch* which is maintained at a high temperature and serves the same purpose as the roof of the Dutch oven. The secondary air is usually admitted with the coal under the arch; and the grate bars are moved in such manner as to carry the fuel bed constantly toward the ash dump at the bottom. While the entering "green" coal is approaching the arch and the hotter portion of the fuel bed, a progressive distillation of the volatile matter occurs. The gases evolved and the secondary air mix and flow under the heated arch, which deflects them so that they pass over the hotter portion of the fuel bed. Thus the conditions are excellent for burning the volatile content of the fuel provided the combustion space has sufficient volume. Stokers of this type ordinarily operate with natural draft, but this is sometimes supplemented by induced or forced draft.

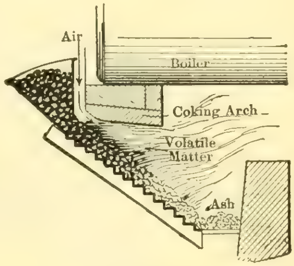

Fig. 564.

(**b**) A typical arrangement of **inclined front-feed stoker** with inclined grate is shown diagrammatically in Fig. 565. It has a hopper, coal-pusher feeding-device, dead plate, coking arch, and air inlet under the latter. The grate bars are stepped and inclined, and they are mechanically oscillated, or reciprocated, in such way as to cause the bed of coal to gradually descend. The rapidity and amplitude of motion of the pushers and grates can be so regulated that the coal is just burned out by the time it arrives at the bottom of the grate. (If combustion is completed before this point is reached cold air may force its way through the thin layer of ashes at the end of the grate and thus reduce the efficiency of the unit; and if not completed some unburned fuel may be lost with the refuse.) The ashes and clinker become deposited on the ash table, which is dumped by hand from time to time. When the ash table is tilted a guard is brought into position (as is shown at (*a*) in the figure) to keep the fuel bed from avalanching down and being dumped at the same time. In the figure, the upper reciprocating grate bars are hollow and the air, which is injected into their interiors by steam jets S, issues through openings in the risers of the steps. The lower grates rock or oscillate about the trunnions shown in (*b*) and have replaceable bars. There are, of course, many other designs and arrangements of front-feed stokers.

(c) A typical arrangement of **double inclined side-feed stoker** is shown diagrammatically in Fig. 566. Coal is fed into the magazines from above, or through doors (*a*) in the front, and is pushed by some suitable mechanism, to the coking plate at the top of the inclined grates. The whole bed of fuel is covered by a fire-brick roof, either flat or arched (as shown), and the secondary air, which is heated by passing over the arch, is discharged into the furnace just above the entering coal. The coal gradually descends on the inclined grate bars, the alternate ones of which are constantly moving. The ash and clinkers are crushed by rotating (or oscillating) grinders located at the bottom of the grates. Some grinders are made hollow and are connected to the draft in such a way as to cause cold air to pass through them to prevent overheating. When clinkering coal is used, steam (from the stoker engine, if there is one) is sometimes discharged through the bed of ashes to reduce the amount of clinker and to make crushing easier. The position of the furnace with respect to the boiler and the arrangement of supplementary combustion space and baffles are much the same as for the Dutch oven.

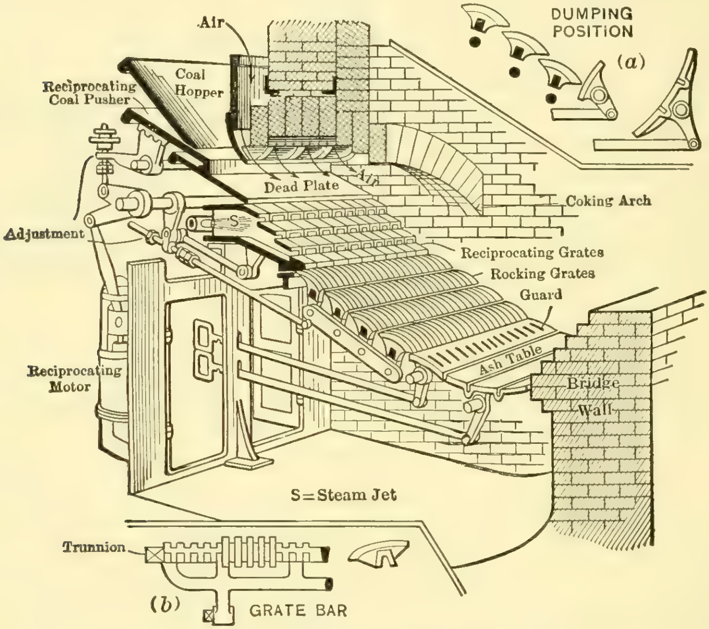

Fig. 565.—Inclined Front-Feed Stoker.

The advantageous features of this type of stoker are the large *coking spaces,* the ample *coking arch,* and the voluminous *combustion chamber.* These stokers operate successfully with both uniform and variable loads and under a great variety of conditions. With some of these stokers, when certain types of coal are used, there is difficulty in getting rid of the ash and clinker, and avalanching may occur if operated at very high rates of combustion.

(**d**) *Inclined overfeed stokers* can be used for burning most any type of coal from lignite to anthracite, and also for sawdust, tanbark, hogged fuel, etc. They are used, however, mainly for high-volatile and high-ash, mid-western coals. The average rate of combustion with free burning coals is from 20 to 35 lb. per sq. ft. of oblique grate surface per hour, and the maximum with forcing is about 50 lb. With caking coal the maximum is about 35 lb. The horizontal projected area is about 70 per cent of the oblique area of the grate. Usually with these stokers more attention and labor is required to keep the fuel bed open, and to take care of the ashes and clinker and the sifting of fine coal, than is necessary with other types. They are not used with large boilers which are operated at high loads.

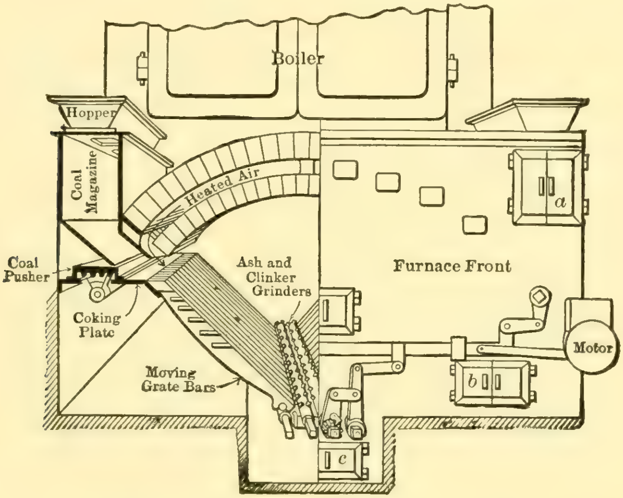

Fig. 566.—Double Inclined Side-Feed Stoker.

412. Chain- and Traveling-Grate Stokers—General.—(**a**) Stokers of this type are basically endless coal conveyors which continuously receive fresh fuel at the front, convey it through the various combustion regions while maintaining the fuel bed *undisturbed,* and automatically discharge the residue at the rear. As the coal moves into and through the furnace its moisture is first vaporized, next the volatiles are distilled, then the fixed carbon is burned, and finally the ash is disposed of, the cleaned grate bars or links returning underneath to the front to repeat the process, and while doing so being cooled by the inflowing air. Some stokers of this type are designed to operate primarily only with *natural draft,* and others to use *forced draft.* All are particularly suitable for **burning non-caking coal,** which may be clinkering (and therefore needs an undisturbed bed) and high in ash, for which the continuous discharge is desirable. An ash content of at least 7 per cent is necessary to provide an insulating layer over the rear of the grate to protect this part from the furnace heat. Two outstanding characteristics of these stokers are: (1) *highly volatile* coals can be easily burned on them without smoke, either with natural or forced draft, *provided* they are supplemented by properly designed arches and combustion space; and (2) *non-volatile* small-size anthracite and coke braize (breeze) are burned satisfactorily on special designs of the forced-draft type, *provided* (again) adequate arches are used. In some early forms there was considerable sifting of fine coal between the stoker sides and the adjacent walls of the furnace, and cold air leaked past the sides, back end and through the

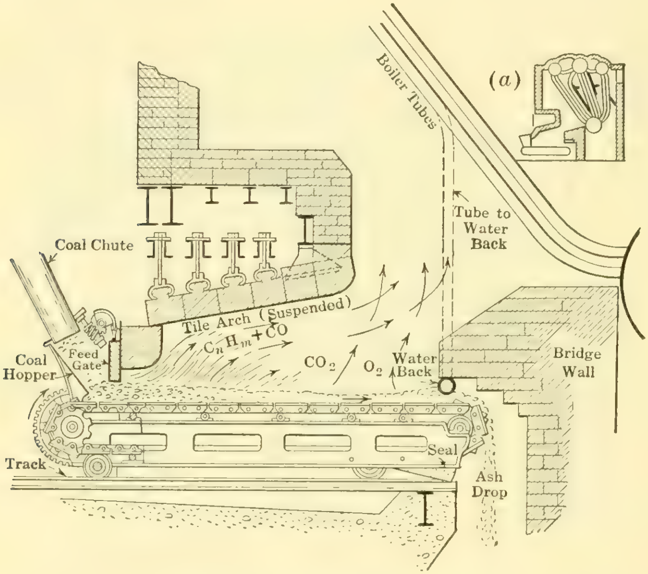

FIG. 567.—Natural-Draft Chain-Grate Stoker with Furnace Suitable for Mid-western Coal.

rear portion of the grate; but in modern designs these faults have been largely or wholly overcome. Stokers of this type, while relatively somewhat expensive, operate with little attention and low maintenance cost; but, unless used with artificial draft, they are less suitable for meeting sudden, heavy variations in load, particularly if the changes cannot be foreseen, than are underfeed stokers or powdered-coal burners, yet to be considered.

(**b**) **Chain grates,** see Fig. 567, are made up of numerous small links, forming endless chains which are carried on suitable rolls or skids and are driven by sprockets at speeds which may be varied to suit the load, fuel, thickness of bed, draft, and excess air. In the arrangement shown, the relative motion which occurs between the interlocking link-ends when the chain passes around the rear guide shears loose any adhering clinker, thereby automatically cleaning the grate. Chain grates are, of course, **traveling grates,** but the latter term is often reserved for the arrangement in which there is a series of crossbars (carrier bars) which are carried at each end by continuous chains at the sides of the stoker and which are covered by abutting, interlocking or overlapping grate-surfaces, made up of many small elements, or keys, as in Fig. 568. In the usual commercial designs the links or keys can be replaced without stopping the stoker, the movement of which is very slow. The various designs of natural-draft and forced-draft chain- and traveling-grates have many features in common.

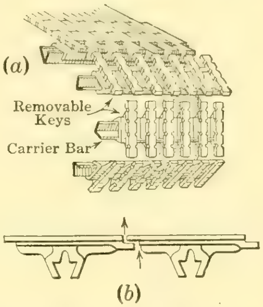

Fig. 568.—Carrier Bars and Keys.

(**c**) With all such stokers the **air leakage** around the grate should be minimized by using adjustable ledge plates to seal the gaps between the sides of the stoker and the furnace wall; the ash pit should be tight so as to reduce infiltration at the back end of the stoker; and there should be a well-fitted damper at the rear between the upper and lower runs and also a seal below the latter. Often an overhanging bridge wall has a **water back** set close to the grate (Fig. 567) to compress the fuel bed, thus increasing the density of the rear portion so that there is less air infiltration at this point; this back retards the upper portion of the bed, giving a better opportunity for the fuel to burn out, thus reducing the amount of fuel discharged with the ash; it provides a water-cooled surface to which clinker will not adhere; and it protects the bridge wall. Often it is necessary to install **side-wall water boxes,** or to make other provision, which will prevent clinker from building up on the adjacent furnace walls and then scraping the sides of the stoker clean, thereby increasing the air leakage.

(**d**) The fuel beds on chain grate stokers are thick and compact at the front and thin and porous at the rear; hence, unless secondary air is provided, there is a deficiency of oxygen over the front of the grate, and there is always an excessive amount over the back part. It is therefore desirable to mix the unburnt gases from the front with the gases rich in oxygen at the rear, so as to ensure complete combustion without using an excessive amount of air. **Arches** *over the fire* may be arranged (1) to bring about this mixing, and also (2) to maintain sufficient temperature to support the combustion of these gases, (3) to radiate to the front of the fuel bed heat received from the hotter portions of the bed so as to expedite the distillation of the volatile matter and the ignition of the entering fuel, and (4) to retain in the furnace much of the "fly" fuel which otherwise is carried up by the gases when a strong draft is used. With high rates of combustion, it is usually desirable to introduce *secondary air* under the front of the arch, preferably by means of a fan (as in Fig. 571) rather than by induced draft or by steam jets. The air thus admitted also prolongs the life of the refractory arch. Sometimes dampers are placed under the back part of the stoker so as to reduce the air filtration there when short thin fires are carried.

But, in addition to having the proper arrangement of arches within the furnace, there should also be provided adequate combustion space. The necessary volume is obtained by placing the boiler surface in a suitable location with respect to the grate. Table XLIX gives the absolute minimum and preferred minimum heights of setting of various kinds of boilers to use with the different types of commercial stokers, as established by the Stoker Manufacturers' Association. In special cases, however, it may be desirable to use heights other than those there given.

(**e**) In the **operation** of such stokers, the *thickness* of the fuel bed at the front is regulated by an adjustable gate at the hopper outlet, and when the proper thickness has once been established for the fuel being burned, it should not be altered. The bed should be as thin as can be used and yet have the fuel ignite rapidly and burn out at the proper point. The variation in load is met best by changing the stoker speed and the draft. The commercial designs for the more volatile coals give the best results with the size of coal that will pass through a 1-in. round hole (although screenings up to 2 in. also give satisfactory performance), and the coal should contain about 50 per cent of "fines" so as to give a uniformly dense fuel bed offering correct resistance to the flow of the primary air. **Tempering** such coal by adding moisture is

TABLE XLIX

HEIGHTS OF SETTINGS FOR VARIOUS TYPES OF BOILERS EQUIPPED WITH STOKERS

TYPE OF BOILER	TYPE OF STOKER											
	UNDERFEED				CHAIN GRATE				OVERFEED			
	Multiple Retort		Single Retort		Natural Draft		Forced Draft *		Side Feed		Front Feed	
	Detroit, Frederick, Jones, Riley, Taylor, Westinghouse		Burke, Detroit, Jones, Type E, Westinghouse		B. & W. Burke, Green, Harrington, Illinois		B. & W. Burke, Green, Coxe, Harrington, Illinois		Detroit, Murphy		Huber, Roney	
	Min.	P.M.	Min.	P.M.	Min.	P.M.	Min.	P.M.	Min.	P.M.	Min.	P.M.
WATER TUBE:	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.
Horizontal, all sizes	11 0	13 0	9 0	11 0	10 0	12 0	12 0	14 0	9 0	11 0	9 0	11 0
Inclined (H.M.D.), all sizes	7 6	8 6	6 6	8 6	6 0	8 0	7 0	8 0	5 0	7 0	6 6	8 0
Inclined (V.M.D.), all sizes	6 0	7 0	5 0	7 0	4 0	5 0	6 0	8 0	3 6	5 0	4 0	5 6
Vertical (H.M.D.) all sizes	3 6	5 0	3 6	5 0	3 6	4 6	4 0	5 0	3 0	4 0	3 6	5 0
Vertical (V.M.D.), 1500 sq. ft.	4 6	5 0	4 6	5 0	4 6	5 0	5 0	5 6	3 3	......	3 6	4 6
Vertical (V.M.D.), 2500 sq. ft.	5 6	6 0	5 6	6 0	4 6	5 0	5 0	5 6	3 3	......	3 6	4 6
Vertical (V.M.D.), 5000 sq. ft.	6 0	6 6	6 0	6 6	4 6	5 0	6 0	6 6	3 3	......	3 6	4 6
HORIZONTAL RETURN TUBULAR:												
72 in.	8 0	10 0	7 6	8 6	7 0	8 0	8 0	10 0	7 0	8 0	6 0	8 0
84 in.	8 0	10 0	7 6	8 6	7 0	8 0	8 0	10 0	7 0	9 0	6 0	8 0

DEFINITIONS OF SETTING HEIGHTS

TYPE OF BOILER	MEASUREMENT OF HEIGHT
WATER TUBE:	
Horizontal tubes (Nominally so)	Floor line to bottom of header above stoker.
Inclined tubes with H.M.D.	Floor line to center of mud drum.
Inclined tubes with V.M.D.	Floor line to top of mud drum.
Vertical tubes with H.M.D.	Floor line to center of mud drum.
Vertical tubes with V.M.D.	Floor line to top of mud drum.
HORIZONTAL RETURN TUBULAR	Floor line to under side of shell.

ABBREVIATIONS:

H.M.D. = Horizontal Mud Drum.
V.M.D. = Vertical Mud Drum.
Min. = Absolute Minimum.
P.M. = Preferred Minimum, i.e., the minimum heights recommended.

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* When burning coke breeze and anthracite fines, the setting heights indicated should be materially increased to provide for proper arch and furnace design.

often practiced to improve the physical behavior of the fuel rather than for its effect on the chemical process. Adding about 3 per cent of moisture, by wetting, decreases the sifting of the coal through the grate, and results in fewer blowholes in the bed; and this reduces the excess air, improves the quality of the ash, and betters the ignition, yet the evaporation of this small quantity of water causes only about 0.3 per cent loss of the heating value of the coal.

While the various kinds of chain- and traveling-grate stokers have much in common, there are many important differences between them. In each application the general design, size and proportions of such stokers should be selected with respect to the kind of draft and fuel used, and to the other special conditions to be encountered; and the furnace arrangement should also be given very careful consideration.

413. Natural-Draft Chain-Grate Stoker Settings.—(a) Fig. 567 shows a typical natural-draft chain grate, with one of the more common arrangements of furnace. Stokers of this type are best suited for free-burning, high-volatile bituminous and sub-bituminous coals and lignites. When used with chimney draft only, the arch height at the front that has given the best results is about 36 in. above the chain; and the roof is pitched slightly upward towards the rear. The principal factors to be considered in designing the arch are (1) the combustion rate, (2) the percentage of volatile combustible in the fuel, (3) the calorific value of the fuel, and (4) the stoker length. Obviously, the higher the combustion rate the greater must be the provision for igniting the coal. Coals of low grade require longer arches than those of high grade. Figure 569, due to T. A. Marsh,[9] gives the arch lengths usually suitable for the various ignition rates possible with a variety of bituminous coals containing at least 25 per cent of volatile combustible. If l is the length of the active portion of the grate, and w lb. of coal are burned per square foot of G. S. per hour, then the ignition rate is $I = w \times l$, pounds of coal ignited per hour per foot of grate width. Knowing I and the calorific value of the fuel, in B.t.u. per pound, the length of the arch in feet can be obtained from this chart.

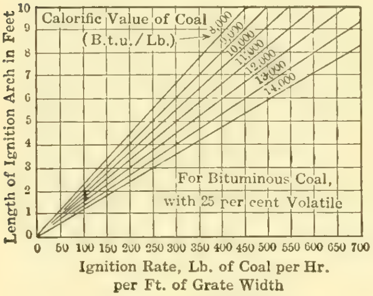

FIG. 569.—Arch Lengths.

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[9] Electrical World, Oct. 23, 1920.

Furnace arrangements other than that shown in Fig. 567 are often used with these natural-draft stokers. Three are shown in Fig. 570, all providing a good degree of turbulence for the burning gases. In the arrangement shown at (A) the mixing is produced partly by the stepped-up arch; and in those shown at (B) and (C) it is due to the increase in velocity in the narrow outlets of the furnaces. The minimum heights at which the different types of boilers should be set with these stokers are given in Table XLIX.

(b) The **combustion rates** usual and most efficient with natural-draft chain grates, when using the free-burning fuels for which these stokers are best suited, are from 20 to 30 lb. of coal per sq. ft. of grate surface per hour. The maximum rate is generally from 38 to 40 lb.; and fairly good results are obtainable with rates as low as 5 lb. The draft needed through the fuel bed is approximately $\frac{1}{10}$ in. of H_2O for each 10 lb. of coal burned per hr. per sq. ft., up to combustion rates of 35. Above this rate the draft needs to be somewhat higher.

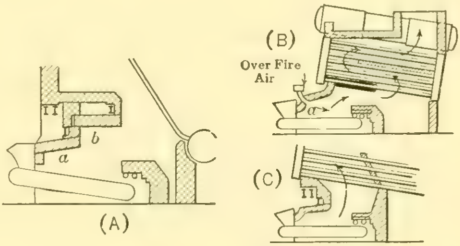

Fig. 570.

With proper operation, the monthly efficiencies of furnace and boiler (without economizer or air preheater) should average slightly above 70 per cent; the CO_2 at the boiler outlet should be 12 or 13 per cent; and the ash will contain from 15 to 25 per cent combustible, with firing rates from 25 to 40 lb. per sq. ft. per hour.[10]

414. Forced-Draft Traveling Grates—General.—Traveling grates for use with *forced draft* were first developed for burning small-size anthracite, and they are practically the only type of stoker suitable for use with this type of fuel, and for culm, river coal and coke braize. Applications were made later to a wide range of other fuels, including semi-anthracite, free-burning bituminous and sub-bituminous coals and lignite. They resemble the natural-draft arrangement of stokers of this type, except that under the active length of the upper strand of the chain they are provided with a series of transverse, independent, forced-draft compartments, each with a separate control for regulating the air pressure under the portion of the grate served by it. In some constructions the compartments consist of boxes which have air-tight sides and bottoms and which are located entirely above the lower chain; while in others, the compartments are formed by vertical transverse plates placed between the upper and lower passes of the chain, the bottom seal being below the lower strand. In the latter construction, the bottoms of the division walls have sealing plates which ride on the lower run of the chain, and the chain itself either rides directly on a floor plate or has sealing plates below it, with ash or sand filling between. Provision should be made also for removing the siftings from the fuel bed and for preventing them from being blown into the boiler room.

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[10] For further discussion, see: "Traveling-grate Stokers," by T. A. Marsh in Power, Feb. 21, March 6, 1921; and "Improving Boiler Room Operation," by Otto deLorenzi, in Combustion, Aug., 1928, p. 84.

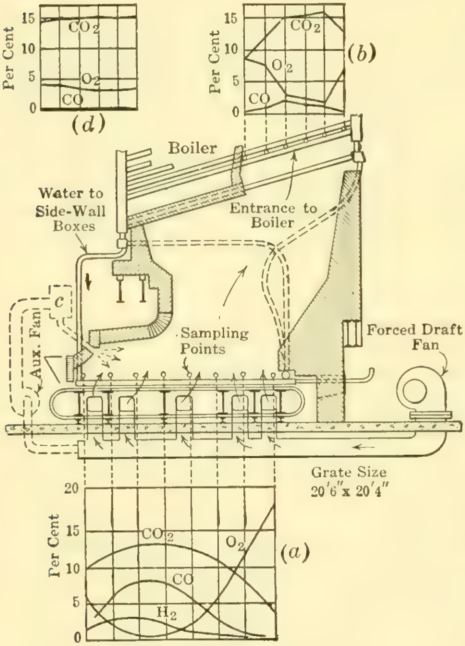

FIG. 571.—Typical Forced-draft Chain Grate Installation, and Gas Analyses Over the Fuel Bed and at Entrance to the First Pass.

As the conditions encountered in burning the more volatile coals are quite different from those with the less volatile ones, the two fields of application will be considered separately.

415. Forced-Draft Chain-Grate Stokers and Furnaces Used with Bituminous Coals and Lignite.—(a) With forced-draft, this type of stoker not only can consume the free-burning coals at greater rates per unit area of surface than is possible with natural draft operation, but also the efficiency curves are both higher and flatter; further, due to the " boiling " or stirring of the fuel bed by the action of the forced draft, these stokers can also be used with some caking and coking coals which could not be used with this type when supplied with natural draft. These forced-draft stokers have been built in sizes as large as 700 sq. ft. of surface, the air being supplied from both sides.

(b) **Furnaces** for these stokers require less extensive arches for a given rate of ignition than are necessary with natural draft, and the arches are set higher to avoid the erosive action of the draft. Water backs are generally desirable, and side-wall water boxes or their equivalent, are almost a necessity. Figure 571 shows diagrammatically one arrangement of furnace for burning Illinois high-volatile coal, the analysis of the gases at different points over the length of furnace and over the entrance to the boiler tubes being shown at (*a*) and (*b*) respectively for operation without the use of secondary air. With the proper introduction of air under the arch a uniformly high percentage of CO_2 is obtained across the furnace outlet (as shown at (*d*) in the figure), the efficiency is improved, smoke is eliminated, and the length of flame, formation of soot and "birdnesting" in the tubes of the first pass are reduced. The curves in Fig. 572 indicate the variation of the combined efficiency of furnace and boiler with different mid-western coals and with various combustion rates; and those in Fig. 573 illustrate the relation between the air pressures in the various stoker compartments when the coal is burned at high rates with bed-thicknesses from 5 to 6 in.,—except, for the semi-anthracite, the thickness is from $3\frac{1}{2}$ to $4\frac{1}{2}$ in. In each case the best air pressures should be determined by test.

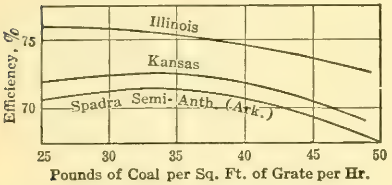

Fig. 572.—Typical Efficiency Variations with Mid-western Coals.

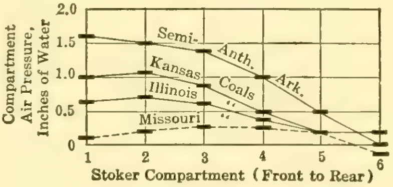

Fig. 573.—Typical Relations Between Compartment Pressures.

Arches placed over the rear end of the stoker and extending forward to within a few feet of the front have resulted in a material increase in capacity; and a combination of both front and rear arches have shown advantages, particularly when a variety of coals must be burned. The rapid destruction of brickwork due to high rates of combustion are avoided by using partly or wholly water-cooled arches and side walls. Fig. 574 shows one double-arch arrangement with suspended replaceable refractory blocks—a setting suitable for burning Illinois slack screenings or crushed run of mine; and Fig. 575 shows arches made up of plain boiler tubes with refractory tile between, the side walls of the furnace also being water cooled, but with a refractory covering from the grate to as high as the throat. In both of these settings secondary air is admitted through the front.

(c) The **combustion rates** giving the best results when bituminous coal is burned on these grates with forced-draft is from 30 to 40 lb. per sq. ft. of surface per hour; but such coals, with from 10 to 25 per cent ash, can be burned readily at a maximum rate of 50 lb., and occasionally at 60 lb. per sq. ft. The average maximum continuous rate of a large number of users in 1927 was found by the National Electric Light Assoc. to be 43.5 lb. per sq. ft. per hr., and the average stoker maintenance cost was 4 cents per ton of coal burned. Variation in rate should be made by adjusting the stoker speed and the main air pressure, without altering the dampers on the compartments, except, when the rate is very low, it may be necessary to carry a very short fire, and then the rear compartments are closed.

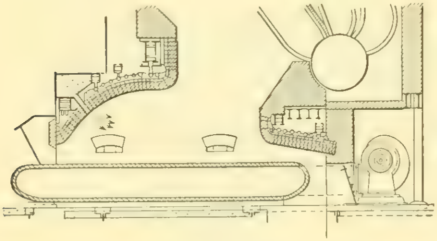

Fig. 574.—Double Suspended Arch Setting.

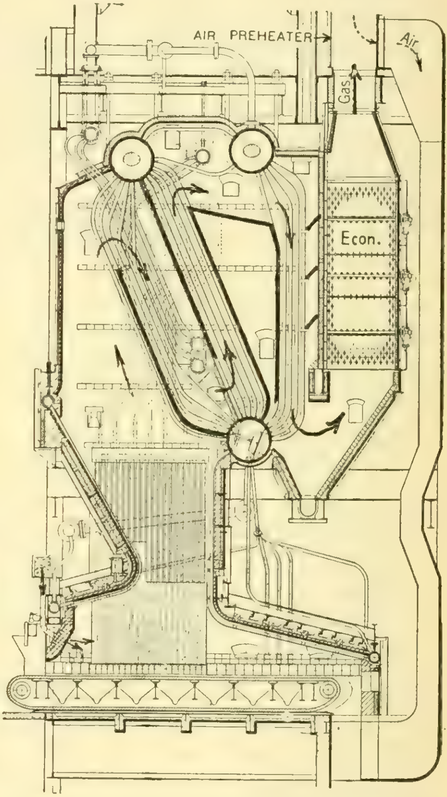

Fig. 575.—Water-cooled Arches and Side Walls.

(d) **Operating results** with these forced draft stokers should show monthly combined furnace and boiler efficiencies (gross) of from 70 to 77 per cent, depending on the fuel and combustion rates used; the CO_2 at boiler outlet should be between 12 and 15 per cent; and the refuse should not contain more than 10 to 20 per cent of combustible. Units with economizer and/or air preheaters will have still higher efficiencies.

Figs. 576 and 577 give the results of a series of tests of a steam-generating unit having a chain grate stoker 20.5 ft. long and 24 ft. wide with furnace arranged much like that in Fig. 571, except that the boiler is set higher and the full length of the first row of tubes is open to the furnace. The areas of the heat-transmitting surfaces, in square feet, are as follows: boiler, 15,676; superheater, 2500; economizer, 11,054; and preheater, about 10,000. The temperature of the air entering the stoker from the air preheater ranged during the tests from about 170 deg. fahr. under low capacity to 225 deg. at high rates of operation. The coal used was Illinois slack, having about 18 per cent moisture, $12\frac{1}{2}$ per cent ash, and a higher heating value of 10,000 B.t.u. per lb.[11] The auxiliary energy is not given in the test data hence the net efficiency of the unit is lacking. Tests of a similar unit having water-cooled furnace walls gave lower efficiencies, probably due to a too extensive application of the water cooling in the arches.[12]

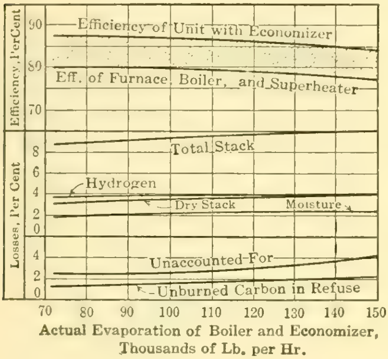

FIG. 576.

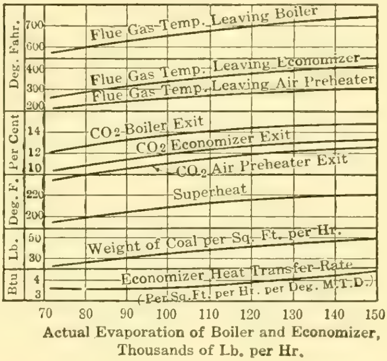

FIG. 577.

416. Forced-Draft Traveling Grates and Furnaces for Burning Low-Volatile Coals.—(a) For *burning small-size anthracite* the air-openings in traveling grates must be so fine as to be mere slits, or else the grate keys must have overlapping ledges so as to prevent sifting. With this fuel and the other low-volatile kinds, the arch arrangement must be such as to ensure the rapid ignition of the entering fuel by means of heat radiated from hot refractory surfaces. The primary kindling must be obtained before fuel reaches the first air compartment, and from there on the ignition then should become accelerated so as to extend through the entire depth of the bed in the first 3 or 4 ft. of travel.

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[11] Proceedings N.E.L.A., 1927, p. 1037.

[12] Proceedings N.E.L.A., 1928, p. 1201.

From that distance onward the ash begins to form and gradually thickens until the fuel is finally burned out. To ensure constant and positive ignition, the arrangement of arch and setting of furnace and boiler must be such that the entering fuel "sees" the hot refractory roof through a greater angle, as *A* in Fig. 578, than that, such as *B*, through which it sees any relatively cold water-heating surface that may be exposed to it. For igniting moist or low-volatile coals quickly and to the full depth of the bed, one manufacturer places a small suction compartment under the front of the stoker so that some of the hot furnace gases are drawn down through the green coal as it enters.

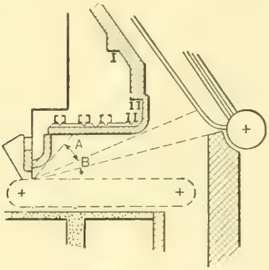

Fig. 578.

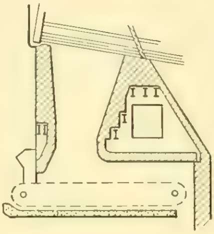

Fig. 579.

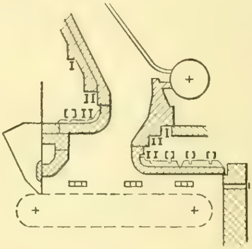

Fig. 580.

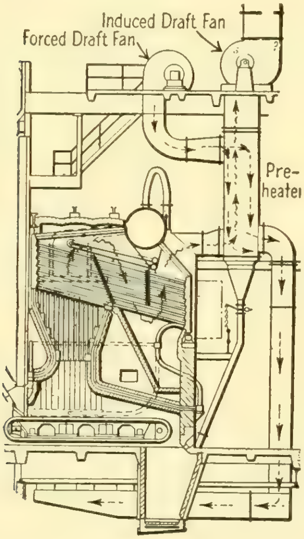

Fig. 581.

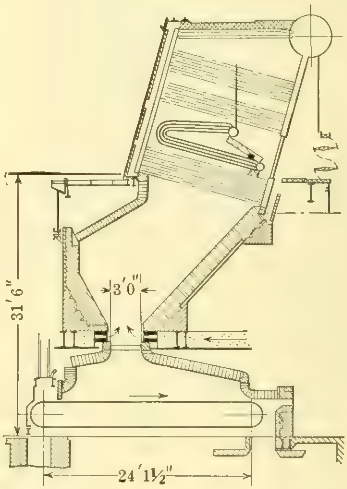

Fig. 582.

With the single front-arch construction, Fig. 578, the gases tend to stratify, the refuse may contain a high percentage of combustible, and there may be difficulty from "fly ash" carried by the gases. These objectionable features may be overcome, and better conditions as to CO_2 are obtained, by the proper introduction of overhead air, as in Fig. 571. The rear arch furnace, Fig. 579, was developed to surmount the same difficulties; but in many installations both front and rear arches are used, as in Figs. 580 and 581. The arrangement can be such that the rich gases moving with *low* velocity from the front of the bed, where the furnace volume is large, and the lean gases (lean in combustible but rich in oxygen) coming from under the rear arch with relatively *high* velocity mix with considerable turbulence in a narrow throat, the combustion being completed in the upper combustion chamber before the gases reach the heating surfaces. Jets of secondary air may be introduced, as in Fig. 582, at high velocity at the constriction to aid in the process. Such expedients raise the efficiencies, reduce the trouble from fly ash, and enable the cheaper grades of coal which contain large proportions of "fines" to be burned more economically.[13]

(b) In **operating** *forced-draft stokers* of this type with the low-volatile fuels, the smaller the percentage of "fines" or undersize particles in the fuel, the higher becomes the efficiency obtainable. The best results are obtained by using thin fuel beds (2½ to 5 in. with fine anthracite) with fast grate speeds, and air pressures that will make the fuel bed dance slightly, giving a white appearance within the furnace. Under high rates of operation with front arch arrangements the air pressure should be about 2 in. of water in the first compartment and should decrease quite uniformly to almost zero in the last compartment; whereas with rear arch arrangements the pressure at the front and rear should be from ¼ to ½ in., with a maximum of about 2 in. at two-thirds the distance from front to rear. The best pressure for each compartment under each loading must be established by actual test. Change in combustion rate is made by varying the stoker speed and air pressure, without altering the thickness of the fuel bed.

(c) **Combustion rates** with *anthracite* or *coke braize* average from 30 to 38 lb. of coal per sq. ft. of grate per hour, using forced draft pressures of from 1.5 to 2 in. H_2O. The maximum rates are from 50 to 55, and the minimum are from 10 to 12 lb. per sq. ft. per hour. Combined boiler and furnace efficiencies of from 72 to 76 per cent are obtainable.

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[13] Figs. 578 and 580 show respectively the original and latest furnace arrangements in the Cornell University central heating plant which burns anthracite Buckwheat No. 3. Fig. 581 shows a furnace in the Hunts Point Plant of the Consolidated Gas Co., N. Y. City, and is designed for burning coke braize, the water-tube arches being faced with refractory (Power, Sept. 6, 1927, p. 350). The arrangement in Fig. 582 is for burning anthracite culm in the Stanton Base Load Plant (Power, Aug. 30, 1927, p. 213 and N.E.L.A. Pub. 068, 1921).

(Suggested specifications for chain grate stokers are given in the Proceedings of the National Electric Light Association, 1927, p. 847.)

417. Underfeed Stokers.—(a) Underfeed stokers are particularly suited for burning bituminous and semi-bituminous, caking or free-burning coals, and can be operated to produce high rates of combustion, especially when the ash content of the fuel is small and of such character that it does not fuse below 2,400 deg. fahr. They are also used with other kinds of coal, having been applied for burning those from every field in the United States. Culm, coke braize and small-size anthracite can be used successfully with them when mixed with bituminous coal.

Stokers of this type operate on the gas-producer principle, using very thick fuel beds. The fuel is fed into them automatically and progressively from *underneath* the fire, gradually being forced upward into the active zone above. The air for combustion is injected, under pressure and in many small jets, into the bed at a level just below that at which combustion takes place. Thus, the hot fire is on the top of the bed and forms an incandescent arch over the fuel which approaches it from below. As the green coal rises and becomes heated it decomposes and chars, forms new solids, and sets free the gases and vapors. The volatile matter ascending is thoroughly mixed with the injected air before entering the combustion zone. Since the volatile matter passes through the incandescent portion of the bed there is practically no opportunity for soot or smoke to form. The porous semi-coke formed has a high order of combustibility. The air jets constantly sweep away the products of combustion from the surface of the porous solid matter under reaction, and rapidly bring to those surfaces fresh supplies of oxygen, thus making possible high rates of combustion. The solid material on the top of the fuel bed is largely ash which, if it fuses at low temperature, tends to form clinker. The action of the stoker is such, however, that there is a slight relative motion between different parts of the bed so as to prevent caking of the clinker and also that of the fuel. The surface residue is automatically worked to the side or rear of the bed for disposal. Arches are not necessary with these stokers, hence a considerable portion of the heat evolved is transmitted radiantly from the red-hot or incandescent surface of the fuel bed directly to the exposed heat-absorbing surfaces of the boiler and water-walls (if any). Under these conditions the stack temperatures remain low even when the unit is carrying high overloads.

There are several typical arrangements of underfeed stokers, each having its special field of application. Usually, however, the fuel is placed in large *hoppers* on the front of the furnace, and from there is fed by reciprocating *rams* into the bottoms of horizontal or slightly inclined trough-like *retorts*, along the upper edges of which are located rows of *tuyeres* for the admission of air. The stokers may have single- or multiple-retorts, supplementary overfeed grates, and various arrangements for the disposal of the ash and for supplying the air. Always there must be a forced draft fan and a source of power to drive it and the stoker, and power may be needed also for operating the ash-removal mechanism.

The various arrangements of underfeed stokers fall into the following classes: (*A*) Simple, single-retort stokers; (*B*) single-retort stokers with lateral grates; (*C*) multiple-retort stokers.

(A) SIMPLE SINGLE-RETORT STOKERS

(**b**) One of the early but still widely used forms of the underfeed stoker is shown in Fig. 583. It has a single horizontal, central retort around the upper edges of which are located replaceable tuyere blocks, and along the bottom of which coal is fed by a ram which is driven by a steam-actuated piston. Supplementary pushers, with adjustable strokes, move in unison with the ram and aid in carrying the coal back into the retort so as to secure a proper distribution. Air from a blower is forced into a sealed pit under the retort. The ashes and clinker fall to the sides onto dead plates, from which they are removed by hand through doors in the furnace front. A screw feed is sometimes substituted for the rams, and other forms of drive may be used. Even with highly volatile

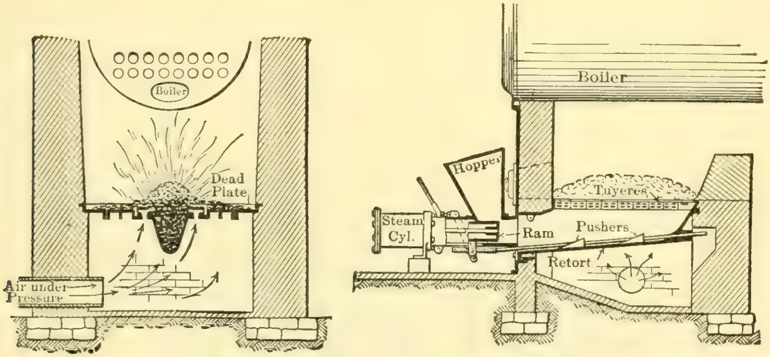

FIG. 583.—Simple Horizontal Single-Retort Stoker.

coals the combustion is complete within a short distance from the surface of the fuel bed if the combustion rate is moderate; hence an extensive combustion space may not be required if high forcing is not used. Stokers of this kind give satisfactory results even when used within cylindrical flues as small as 3 ft. in diameter, such as the corrugated furnace-flues used in some types of internally fired boilers like the Scotch Marine. These simple single-retort stokers are used under boilers ranging in size from 500 to 1500 sq. ft. of heating surface, and their fuel burning capacities are from 800 to 1200 lb. per hour. Sometimes two or three of these stokers are placed in a single large furnace. Pivoted, side *dump plates*, which may be tilted to drop the ashes into pits located below them, may be substituted for the stationary dead plates. The valves controlling the steam cylinder which actuates the ram may be driven in conjunction with the forced draft fan, the speed of which is regulated by the steam pressure; thus, at all times the air and coal may be supplied automatically in correct proportions although the amounts are varied to maintain the steam pressure constant under the fluctuating demands.

(B) SINGLE-RETORT STOKERS WITH LATERAL GRATES

(c) By placing short overfeed grates laterally on one or both sides of the retort, between it and the dump plates, as in Fig. 584, and by providing the extended surface with a suitably-regulated supply of air, the capacity of the furnace, with a *single retort*, can be increased to that needed for boilers having from 1500 to 6000 sq. ft. of heating surface, the weight of fuel burned per hour ranging from 1200 to 9000 lb. according to the size of the stoker used. The lateral grates may be made to move so as to keep the fuel bed porous, and the bars are sometimes made hollow so that some of the combustion air can circulate through them

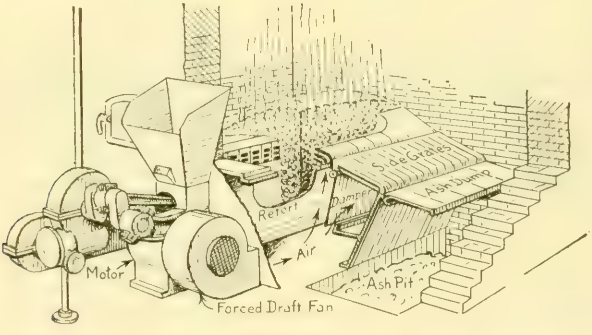

Fig. 584.—Single-Retort Stoker with Lateral Grates.

to keep them cool. The feed ram may be reciprocated by a steam-actuated piston, as in Fig. 583, or by a hydraulic drive; or it may be operated by means of a mechanically-driven crank shaft and connecting rod. In the latter case the motor used to drive the shaft may also operate the forced-draft fan, as in Fig. 584. Both the fuel and the air supply can be regulated automatically as with the simpler form of apparatus discussed in (**b**).

The underfeed stokers so far considered are suitable not only for burning caking and non-caking bituminous coals, but can also be utilized with a wide range of other fuels. With them the boilers can be arranged side by side in continuous batteries since it is not necessary to have access to the sides of these stokers or of their furnaces. The space required in the boiler plant having several units may thereby be reduced over that necessary when access room must be provided on one or both sides of each boiler.

(C) MULTIPLE-RETORT UNDERFEED STOKERS

(**d**) By arranging a *number of underfed retorts side-by-side*, with the necessary tuyeres and air passages between, the active combustion area of the fuel bed may be made to occupy the full width between boiler walls; and by keeping the bed constantly moving from front to rear the refuse can be fed continuously to an ash dump at the back. Such constitutes in the main the arrangement of a multiple-retort type of underfeed stoker, of which the typical elements are illustrated in Fig. 585. The retorts are usually inclined slightly as shown. The air

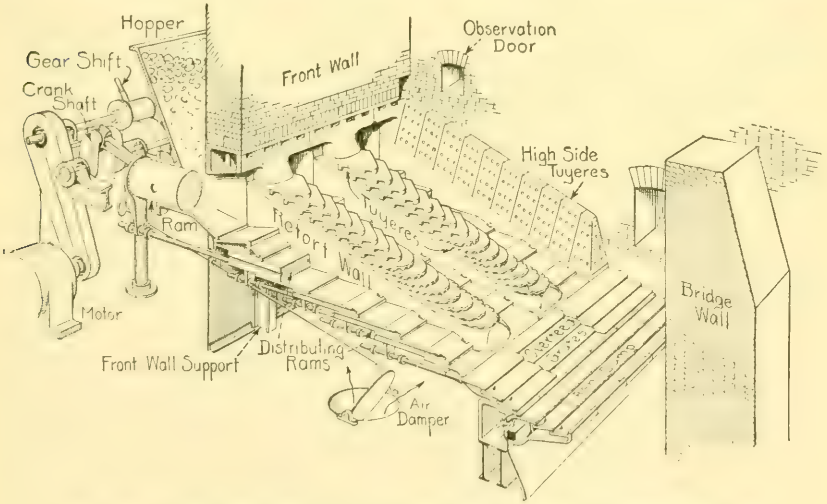

Fig. 585.—Elements of an Inclined Multiple-Retort Underfeed Stoker.

is supplied through openings in replaceable *tuyere-blocks* which overlap shingle-fashion between the retorts. These stokers are built in sizes suitable for furnaces ranging in width from 6 to 28 ft. or more, and in lengths from 8 to 27 ft. or greater; that is, there may be from 3 to 16 or more retorts placed side by side, each retort having from 13 to 69 or more tuyeres in its length.[14]

Compared with other stokers, these can burn coal at much higher rates per square foot of grate area; and, in large units, can occupy a greater proportion, often all, of the area under the boiler. As the energy-absorbing capacity of the heat-transmitting surfaces in a steam-generating unit is generally greater than the heat-releasing capacity of any grate that can ordinarily be incorporated in the unit, less surface for a given rate of steam generation is required in the boilers, economizers, etc., served by stokers of this type, and the cost of such equipment may therefore be less, than when any of the other types of stokers are used; also these underfeed stokers permit making *individual units* that have much greater steam-generating capacities than is possible with other stokers. A very large unit may even have two stokers,—i.e., stokers are placed at each end and both discharge into a common central ash hopper, the firing being done at both the front and rear of the unit.

With the wide range provided with underfeed stokers in the rate of feeding and also in the intensity of draft available, and owing to the rapidity with which adjustments can be made, it is possible to bring the furnace fired with such apparatus very quickly from a banked condition to its full capacity, and to meet wide and rapid changes in load expeditiously.

(e) In addition to the power-driven *feed-ram* or plunger, which can be operated at different speeds, there are also usually provided supplementary *distributing rams* or pushers along the bottom of the retort, as in Fig. 585. These distributors reciprocate in unison with the feed rams, but are adjustable as to motion so that any desired thickness and contour of the fuel bed can be maintained to suit the fuel and operating conditions (see Fig. 586). In some of the arrangements, substantially the entire length of the retort bottom is formed by a series of distributing rams, which can be so adjusted as to make the retort have any desired depth. In one design the inclined retorts, together with the side-wall tuyeres which they carry, are reciprocated, the alternate elements always being moved in opposite directions in adjustable amounts. With all arrangements, the motion of the distributors should be so adjusted as to distribute the coal properly over the retorts, to keep the fuel bed porous and free from clinkers, and to cause the desired rate of movement from front to rear. Stokers for long, high-capacity furnaces are usually provided also with either stationary, reciprocating or undulating *overfeed grates* which extend beyond the ends of the retorts, as in Fig. 585. The active area of the fuel bed may be divided into zones, each with air supply independently regulated either manually or automatically, one arrangement of air passages being shown at 1, 2, 3, 4, and 5 in Fig. 588.

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[14] Some of the largest underfeed stokers are 26.5 ft. wide by 27 ft. long, and are capable of burning 56,000 lb. of coal per hour at maximum rating. They are located in the Hudson Ave. Station of the Brooklyn Edison Co. Manufacturers are ready to build stokers suitable for boilers generating 800,000 lbs., or more, of steam per hour.

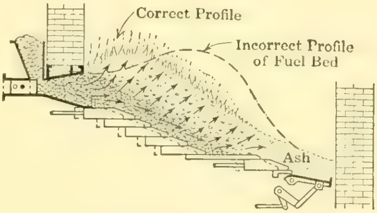

FIG. 586.

(f) The *crank-shaft drive* is by means of variable-speed electric motor, turbine, steam engine or hydraulic motor, usually through a *speed-reducing gear box* which is part of the stoker and which provides two or more speed ratios and a neutral between the speed shaft and the crank shaft. In the larger units the crank shaft is in sections, each of which can be driven at a speed independent of the others. Often the drive is from a line shaft which is located under the stoker, sometimes below the floor, and which may be driven at variable speeds. In some stokers the rams are actuated directly by hydraulic plungers. The stoker mechanism usually contains a *shearing-pin*, or equivalent safety device, to protect important parts from being broken should foreign matter, such as car couplings, spikes, bricks, etc., obstruct the movement of the ram.

(g) Various **ash-discharging devices** are used with stokers of this kind and some are illustrated in Figs. 587 and 588, all being mechanically operated. In the first figure the *dump plate* shown in (*a*), in addition to discharging the ashes, can also be raised to shear off the clinker that accumulates on the bridge wall above it. At (*b*) in the same figure is shown an arrangement with *double dumping grates* which are made hollow so that air may be passed through them, both to cool them and to complete the combustion of any fuel that remains in the ashes on them. In the arrangement shown in (*c*) of this figure the *rocker plates* are reciprocated constantly; and adjacent ones move in opposite directions and continuously agitate, crush and discharge the ash. The hand wheel is used for adjusting the clearance A, or for the complete dumping of the ash. In another common arrangement the ash is discharged by single or double *clinker grinders.* The latter are shown in Fig. 588, at the bottom of a deep ash pocket, the toothed rolls being operated by power either continuously or intermittently. By making the grinder pocket large enough to hold the ash for 12 hours, and by forcing air through the walls of the pocket, the amount of unburned combustible in the refuse finally discharged can be reduced to below 5 per cent of the "ash." If the time of burning out is reduced to but six hours the combustible percentage is about triple this amount; and if but one hour is allowed about half the refuse will be carbon, when the rates of combustion are high. With dump grates the carbon in the ash is often between 15 and 25 per cent.

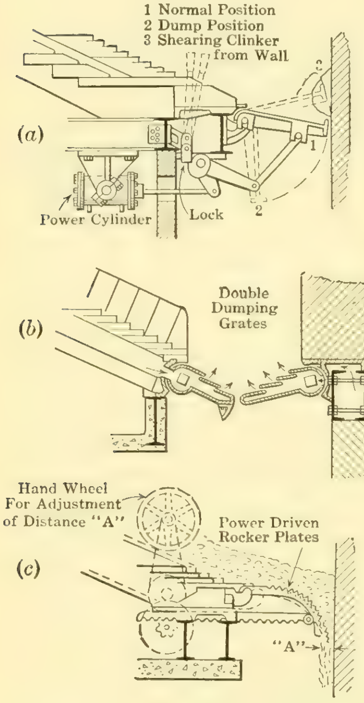
1 Normal Position
2 Dump Position
3 Shearing Clinker from Wall
3
(a)
1
Power Cylinder
Lock
2
(b)
Double Dumping Grates
Hand Wheel For Adjustment of Distance "A"
Power Driven Rocker Plates
(c)
"A"

FIG. 587.—Arrangements of Ash Dumps.

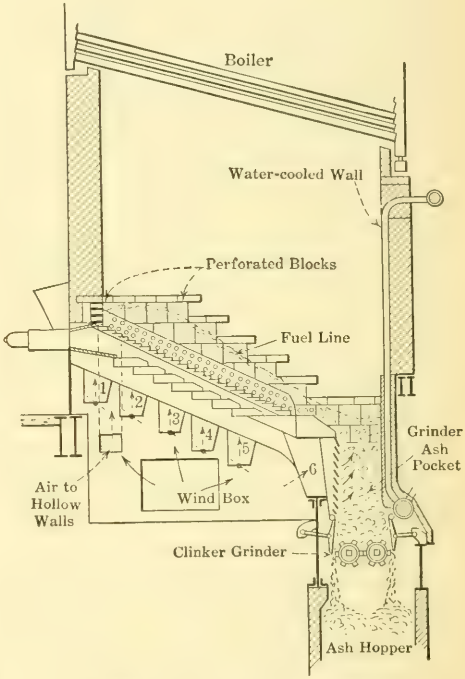

FIG. 588.—Clinker Grinders.

(h) **Furnace walls,** used with underfeed stokers, especially when the rates of operation are to be *intensive,* should be of such materials and construction as will withstand not only the high furnace temperatures but also the erosive action due to the flow over the wall surfaces of large quantities of molten fly-ash which, with some fuels, becomes deposited on them under high forcing. The walls may be made of special grades of fire brick, of silicon carbide blocks (like "carborundum") or of other highly refractory material; they may be made hollow and have the primary air for combustion circulated through them; they may contain hollow perforated blocks through which secondary air is circulated and discharged, as shown in the front and side walls in Fig. 588; or they may consist partly or wholly of refractory- or metallic-surfaces that are water cooled, as in the back wall of the same figure. High, air-cooled side-wall tuyeres, as in Fig. 585, or water-cooled metallic surfaces, as in Fig. 589, may be used to protect the side walls from erosion by the moving bed of fuel and ash, and to prevent the adhesion of molten clinkers. Openings for inspection, and also for the cleaning of refractory walls by means of slice bars, should be provided on at least one side of the furnace; hence the units having these stokers are arranged either singly or in batteries of not more than two, each unit in the latter case having access openings on one side.

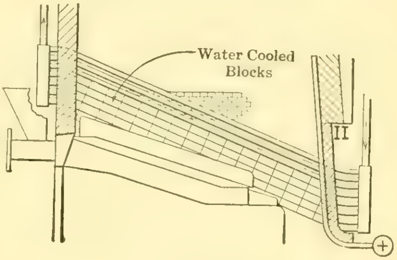

Fig. 589.

(i) The **rates of combustion** with multiple-retort underfeed stokers may be varied from banked operation to 75 lb. of coal per sq. ft. of G. S. per hour, or more, the upper limit depending on the kind of the coal, furnace design, and the intensity of forced draft available. The best operation is usually obtained with rates of from 35 to 45 lb. The average maximum continuous combustion rates of a number of central stations in 1928 was 50 lb. per sq. ft. of G. S. per hour; and the average total stoker maintenance cost was 9.5 cents per ton of coal burned. Combustion rates as high as 89 lb. per sq. ft. have been carried efficiently and satisfactorily over long periods of time. In large stokers over one ton of coal may be burned per retort per hour under normal operation, and in short stokers less than half this rate is obtained. Over short periods of time higher rates are permissible.

(j) The **combustion air** used with stokers of this type can be kept comparatively low in amount, as but little excess air is required. The excess percentage should, of course, be sufficient to ensure complete combustion without discharging CO, and not be so small as to result in a furnace temperature higher than the wall construction will withstand.

The **wind box pressure** used with these stokers is from $\frac{3}{4}$ to 1 in. of water for each 10 lb. of coal burned per sq. ft. of G. S. per hour; but the amount depends on the character of the coal, design of tuyeres, the resistance of passages, etc. Generally, higher pressures are required for low-volatile coals than for high-volatile ones.

When *preheated air* is employed, and temperatures of 300 to 500 deg. fahr. have been adopted in a number of installations, the design and materials used for the stoker should be such that the expansion and growth of metal will not reduce the air passages, or induce prohibitive stresses and distortions. At 600 deg. fahr. air has double the volume it has at 70 deg. fahr., and the sizes of the passages, fans, and fan motors must be proportioned accordingly.

The **duct area** required with cold air is roughly from 1 to 1.3 sq. ft. per retort of moderate size, the velocity of travel being from 2000 to 3000 ft. per min., depending on the rate of operation of the unit.

(**k**) When high rates of combustion are used much *fine cinder* is discharged with the exit gases; hence, in localities where the deposit of this fine dust is objectionable, some form of *cinder eliminator* must be used to clean the gases. Also, soot-blowers or other means must be provided for removing the cinder deposits from the heat-transmitting surfaces of boilers, superheaters, economizers, air preheaters, and flues through which the gases pass; and there must also be means for removing deposits from cinder-catching pockets provided in the settings.

(l) The **power required** to operate underfeed stokers is used (1) to feed and distribute the coal, (2) to operate the ash dumps or clinker grinders, and (3) to supply the required quantity of air at the necessary pressure. An average demand as low as 0.2 hp. per retort for driving the stoker and crusher rolls is claimed for some stokers, but drives of larger size should be furnished to meet extraordinary conditions. Roughly from $\frac{3}{4}$ to 1 hp. are installed per retort burning from 700 to 1100 lb. of coal per hour. Modern stokers of this type require only about 7 kw-hr. per ton of coal for operating the fans and driving motors.

(**m**) **Performance** curves of a large steam-generating unit using a multiple retort underfeed stoker (15 retorts, 33 tuyeres) are shown in Fig. 591.[15] Since the power required to operate the forced and induced draft fans during this test is not available, only the gross efficiency of the unit is given. The small amount of power required to operate the stoker at various loads is as shown. The steam pressure is normally 420 lb. per sq. in. abs. and the temperature 700 deg., while the temperature of the feed water entering the economizer varies from about 190 deg. fahr. at low loads to about 270 at high ones.

[15] Data obtained from Proc. N.E.L.A. 1930, p. 1060.

The unit contained the following heating surfaces:[16]

Boiler (cross-drum type)	23,764 sq. ft.	55.1 per cent
Superheater (3 pass interdeck)	3,037	7.0
Water walls	1,326	3.1
Economizer	14,975	34.8
Total heating surface	43,102 sq. ft.	100 per cent

For a unit containing no preheater the efficiency is seen to be very satisfactory over a wide range of loads. Note that the loss due to the combustible in the refuse is remarkably low even at very high rates of firing; also observe that this stoker was fired up to a rate of nearly 90 lb. of coal per hour per square foot of its surface.

The curve of CO$_2$ percentage indicates that the excess air was kept to very low values, say from about 20 per cent at low loads to 10 per cent at high ones. The resultant high furnace temperatures were permitted because the furnace walls were water cooled.

In operating this unit the induced draft was relatively high compared with the forced draft under the stoker so that the gas pressure in the furnace was slightly below atmospheric.[17]

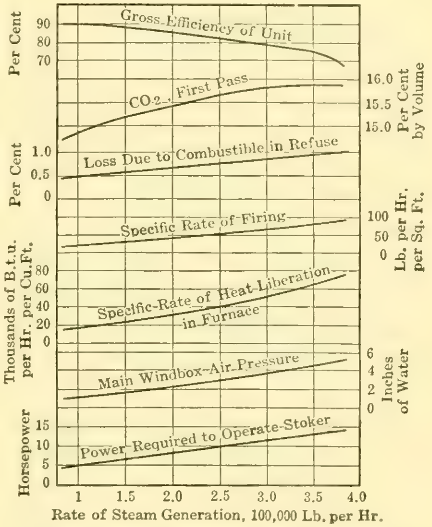

FIG. 591.—Test Results of a Stoker-Fired Steam-Generating Unit at Hudson Avenue.

Some indication of the enormous development that has taken place in stoker-fired units since 1910 is shown by the curves in Fig. 592 based on a more extensive chart prepared by J. G. Worker.[18] The later improvements in efficiencies and the high rates of operation are due not only to better stoker design and selection, but also to improvements in furnace arrangement and to the addition of economizers, air preheaters and water walls.

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[16] See Proc. N.E.L.A., 1930, p. 1060. For test of a later unit, see p. 456.

[17] This method of operation is claimed to increase the efficiency of the unit, to reduce the cost of stoker and furnace maintenance and to lengthen the time the unit can be kept in service. See "Advantages of Thin Fires and High Draft," by Bert Houghton, Power, Sept. 30, 1930.

[18] The Use, Improvement, and Test Results of the Multiple Retort Underfeed Stoker in the United States," by J. G. Worker, Trans. Inst. of Eng'rs and Shipbuilders in Scotland, 1927.

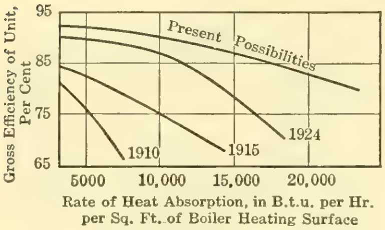

Fig. 592.—Increases in Efficiencies of Stoker-Fired Units Since 1910.

420. Selection of Fuel-Burning Equipment.—(**a**) Stokers, or other types of fuel burners, should be selected with respect to: (1) the characteristics of the fuel to be used; (2) the ability to carry the normal load with high efficiency and to meet the maximum demand that may occur; (3) the flexibility of operation, or ability to meet rapid changes in load; (4) the arrangement desired for the boiler, furnace and other parts of the unit; (5) the ability to use preheated air, when desired; (6) the initial costs involved, including those for the boiler, economizer, air preheater and auxiliary apparatus, and also for the ground and building space occupied; (7) the operating expenses for fuel, labor, maintenance and power to drive all the fuel handling, preparing and burning equipment, including fans; (8) the nuisance from smoke, cinder and fly-ash in the flue gas; (9) the loss from unburned fuel; (10) the loss of heat from the unit when the fire is banked, and the quickness with which the unit can be brought from this condition to full steaming capacity; and (11) the durability of the combustion apparatus and related equipment.

(**b**) The **types of stokers generally most suitable** for the various fuels are given in Table L.[19] This table is only tentative, however, as the art

Table L

Fuel	Preferable Stoker
Anthracite	Traveling forced-draft grate stoker
Coke Breeze	Traveling forced-draft grate stoker
Semi-anthracite	Traveling forced-draft grate stoker
Semi-bituminous (coking)	Underfeed and inclined overfeed stoker
Bituminous (coking)	Underfeed and inclined overfeed stoker
Bituminous (free-burning, ash > 10 or 12%)	Forced or natural-draft traveling grates
Bituminous (free-burning, ash < 10 or 12%)	Traveling grates, underfeed or side inclined*
Sub-bituminous coal	Traveling grates, forced or natural draft
Lignite	Traveling grates, forced or natural draft

* If ash fuses at a temperature below 2,400 deg. fahr., traveling grates are preferable. If the percentage of ash is less than 7, underfeed stokers are preferable.

The term "traveling grates" is used to cover chain grates as well as traveling carrier-bar stokers.

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[19] By T. A. Marsh in "Design and Application of Traveling Grate Stokers," in Power, Feb. 21, 1928, p. 328.

of burning coal is still, to a large extent, in the developmental stage. For example, the use of preheated air now permits fuels high in moisture to be burned by methods which were partly or wholly inapplicable hitherto. The heat thus applied affects the rate of drying, also the coking and clinkering characteristics of the fuel, the temperatures in the furnace and at the boiler exit, the fuel content in the ash, and the efficiency of the unit.

(c) The *interdependence* of the stoker and related parts is important, because the kind and size of stoker may have an influence on the choice of the other equipment and on the size and arrangement of the combined unit, and vice versa. The larger and more effective the heat-transmitting surfaces in the boiler, superheater, economizer and preheater, the smaller can be the stoker and furnace, because of the better combined efficiency. But the larger related equipment, together with the ground and building space occupied, increases the first cost, and this added expense may more than offset the saving in initial cost of the combustion apparatus and in the cost of operation; hence, the financial factors involved in the entire unit must always be considered when making a selection. In small plants the initial cost of equipment per unit of output is usually considerably more than that in larger plants, hence it is often advisable, even at a sacrifice of efficiency, to use in them cheaper and less extensive equipment than would be considered good practice for larger installations.

(d) Suggestions for drawing up *specifications* for underfeed and chain grate stokers, and for preparing instructions for bidders, are given in the Prime Mover Reports in the Proceedings of the N.E.L.A. for 1925 and 1927, respectively. The N.E.L.A. reports each year contain a wealth of additional information on stokers and furnaces and on their operation. Also many reports before the A.S.M.E. and other societies, and many articles in periodicals on this subject, are frequently appearing. The details of different makes of stokers are being constantly improved by the various manufacturers, whose latest catalogues should be consulted; a convenient reference of a general character is the " Condensed Catalog of Mechanical Stokers," compiled by the Stoker Manufacturers Association, Detroit. Some additional references are given on page 599.

CHAPTER XXXII

THE BURNING OF POWDERED COAL

421. Introduction.—(a) Finely divided solid fuel when carried in suspension in air has some of the ideal combustion characteristics possessed by gaseous fuels. Attempts to burn coal in this way started in 1818, but many serious difficulties were encountered. Commercial success was not obtained until about 1895, when powdered coal was applied to firing long cement kilns. Following this date the use of the fuel in this form was extended to many large metallurgical and industrial furnaces. Although the art of preparing and burning the fuel was developed rapidly after this time in these fields, little progress was made in the use of powdered coal under boilers until eventually the discovery was made that a very much larger combustion space must be used than had been the practice with coal burned on grates or mechanical stokers, and that the furnace must be of different design. After extensive experiments conducted in 1920 in its Oneida Street Power Station, the Milwaukee Electric Railway and Light Co. built the first new central station (the " Lakeside ") to rely solely on powdered coal. The plant operated satisfactorily with this fuel, and in the following year was subjected to extensive tests from which many valuable data were obtained. The investigations carried out in the two plants were conducted with the cooperation of the U. S. Bureau of Mines, and the results were reported in the Bureau's Bulletins 223 (1923) and 237 (1925), respectively.[1] Subsequently the use of powdered coal under boilers was rapidly adopted by others.

The arrangement of the Lakeside equipment in 1924 is shown diagrammatically in Fig. 593, which also gives the design and operating data, as presented by M. K. Drewry in his paper entitled " Comparison of Actual Performance and Theoretical Possibilities of the Lakeside Station," Tr. A.S.M.E. 1925, p. 329.

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[1] Also see Bu. of Mines Tech. Paper 316, "Test of a Powdered-Coal Plant, River Mines, Mo." (1923); Bul. 214, "Tests of Marine Boilers"; Bul. 217, "Preparation, Transportation and Combustion of Powdered Coal" (1923); and later Technical Papers and Bulletins of the Bureau.

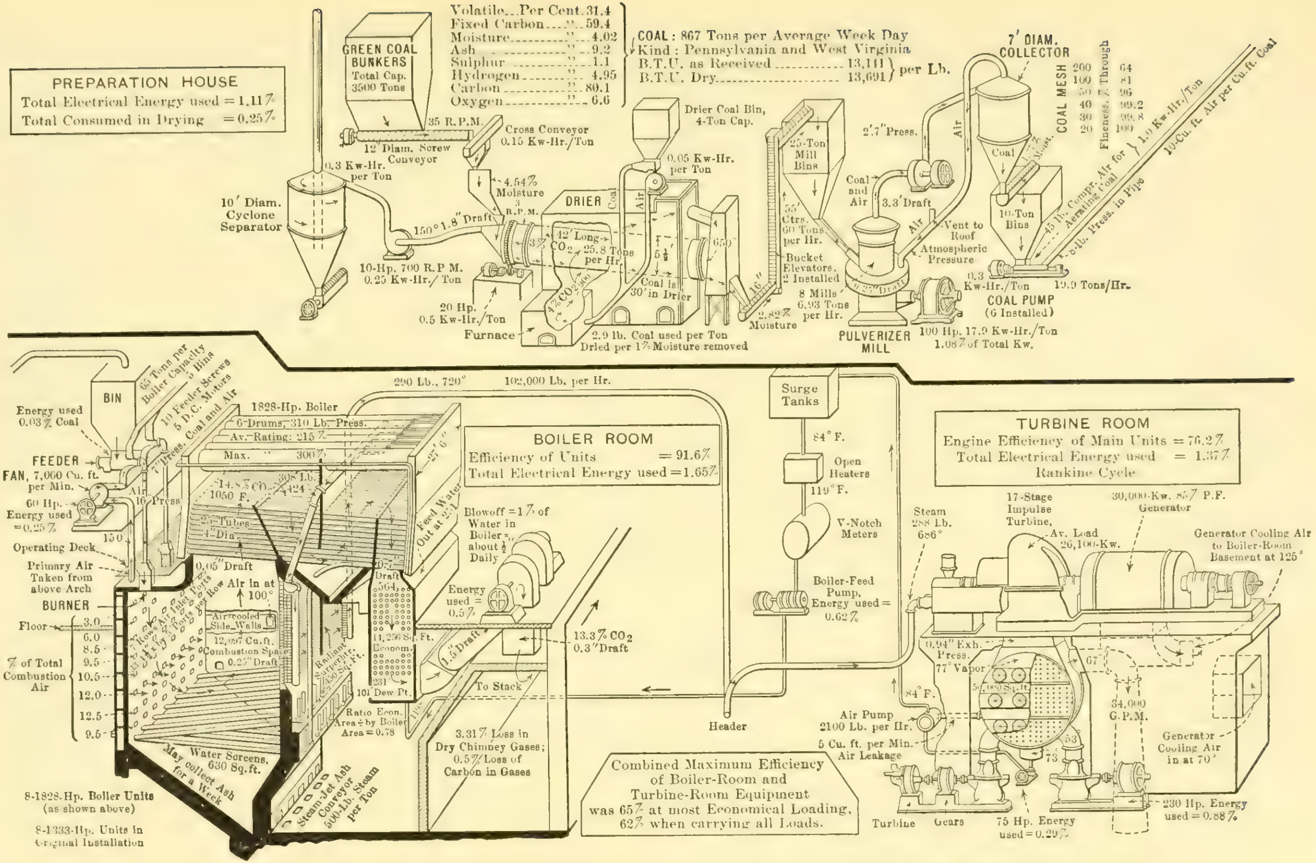
PREPARATION HOUSE
Total Electrical Energy used = 1.11%
Total Consumed in Drying = 0.25%
GREEN COAL BUNKERS
Total Cap. 3500 Tons
Volatile...Per Cent. 31.4
Fixed Carbon..." ..59.4
Moisture..." ..4.02
Ash..." ..9.2
Sulphur..." ..1.1
Hydrogen..." ..4.95
Carbon..." ..80.1
Oxygen..." ..6.6
COAL: 867 Tons per Average Week Day
Kind: Pennsylvania and West Virginia
B.T.U. as Received.....13,111
B.T.U. Dry.....13,691
per Lb.
35 R.P.M.
12' Diam. Screw Conveyor
0.3 Kw-Hr. per Ton
Cross Conveyor
0.15 Kw-Hr./Ton
10' Diam. Cyclone Separator
4.54% Moisture
3 R.P.M.
150° 1.8" Draft
10-Hp. 700 R.P.M.
0.25 Kw-Hr./Ton
DRIER
Coal
Air
12' Long
3%
CO2
25.8 Tons per Hr.
4% CO2
5 1/2
Coal is 30' in Drier
650°
20 Hp.
0.5 Kw-Hr./Ton
Furnace
2.9 lb. Coal used per Ton Dried per 1% Moisture removed
Drier Coal Bin, 4-Ton Cap.
0.05 Kw-Hr. per Ton
16"
2.82% Moisture
25-Ton Mill Bins
55' Ctrs. 60 Tons per Hr.
Bucket Elevators, 2 Installed
8 Mills 6.93 Tons per Hr.
PULVERIZER MILL
Coal and Air
3.3" Draft
2.7" Press.
Air
Vent to Roof
Atmospheric Pressure
100 Hp. 17.9 Kw-Hr./Ton
1.08% of Total Kw.
7' DIAM. COLLECTOR
Coal
10-Ton Bins
0.3 Kw-Hr./Ton
COAL PUMP (6 Installed)
19.9 Tons/Hr.
COAL MESH
Fineness, % Through
200 64
100 81
50 96
40 99.2
30 99.8
20 100
45 lb. Compr. Air for Aerating Coal
1.0 Kw-Hr./Ton
10-Cu. ft. Air per Cu. ft. Coal
5-lb. Press. in Pipe
BIN
65 Tons per Boiler Capacity 5 Bins
10 Feeder Screws 5 D.C. Motors
7" Press. Coal and Air
Energy used 0.03% Coal
FEEDER
FAN, 7,000 Cu. ft. per Min.
60 Hp. Energy used = 0.25%
Air
16" Press
150
Operating Deck
Primary Air Taken from above Arch
BURNER
Floor
% of Total Combustion Air
3.0
6.0
8.5
9.5
10.5
12.0
12.5
9.5
1828-Hp. Boiler
6 Drums, 310 Lb. Press.
Av. Rating: 215%
Max. " 300%
14.8% CO2
1050 F.
308 Lb.
424
25 Tubes 4" Dia.
27' 6"
0.05" Draft
Air in at 100°
Air-cooled Side Walls
12,097 Cu. ft. Combustion Space
0.25" Draft
Rows Air Inlet Ports
Ports per Row
Radiant Superhtr. 1450 Sq. Ft.
0.7" Draft
564
11,256 Sq. Ft. Econom.
231
101 Dew Pt.
Ratio Econ. Area ÷ by Boiler Area = 0.78
Water Screens, 630 Sq. ft.
May collect Ash for a Week
Steam-Jet Ash Conveyor 500-Lb. Steam per Ton
8-1828-Hp. Boiler Units (as shown above)
8-1333-Hp. Units in Original Installation
290 Lb., 720°
102,000 Lb. per Hr.
Feed Water Out at 231
BOILER ROOM
Efficiency of Units = 91.6%
Total Electrical Energy used = 1.65%
Blowoff = 1% of Water in Boiler = about 1/2 Daily
Energy used = 0.5%
1.5" Draft
13.3% CO2
0.3" Draft
To Stack
3.31% Loss in Dry Chimney Gases; 0.5% Loss of Carbon in Gases
Header
Surge Tanks
84° F.
Open Heaters
119° F.
V-Notch Meters
Boiler-Feed Pump, Energy used = 0.62%
Combined Maximum Efficiency of Boiler-Room and Turbine-Room Equipment was 65% at most Economical Loading, 62% when carrying all Loads.
TURBINE ROOM
Engine Efficiency of Main Units = 76.2%
Total Electrical Energy used = 1.37%
Rankine Cycle
Steam 288 Lb. 686°
17-Stage Impulse Turbine,
Av. Load 26,100-Kw.
30,000-Kw. 85% P.F. Generator
Generator Cooling Air to Boiler-Room Basement at 125°
0.94" Exh. Press.
77° Vapor
50,000 Sq. ft.
67
53
73
34,000 G.P.M.
84° F.
Air Pump 2100 Lb. per Hr.
5 Cu. ft. per Min. Air Leakage
Turbine
Gears
75 Hp. Energy used = 0.29%
Generator Cooling Air in at 70°
230 Hp. Energy used = 0.88%

FIG. 593.—Operating Data and Design Figures of the Lakeside Station (Milwaukee, Wis.), 1924.

(**b**) The **principal desirable features** of the use of pulverized fuel are: (1) the possibility of mixing the fuel and air intimately; (2) the consequent feasibility of using only a small amount of excess air—often as low as 20 per cent, vs. 40 per cent or more common with stokers; (3) the ability to burn the fuel at high rates in the furnace area available under a boiler; (4) the flexibility of control which allows quick response to varying demands (there being no massive fuel bed that requires time to heat up or cool down), and which at all times makes it easy to maintain the best conditions of operation even though the load is fluctuating; (5) the practicability of burning a wide variety of fuels, including many of the poorer and cheaper grades and those formerly useless (often a considerable variety of fuels can be used with the same equipment and with little variation in efficiency, especially when the original design is for the worst grade to be used); (6) the absence of moving metallic parts within the furnace, most repairs to apparatus being to external elements; (7) the low cost of maintenance; (8) the ability to use air that is highly preheated; (9) the minimizing of the fuel wasted in starting, stopping and banking and the ability to bring a cold or banked boiler into service quickly; and (10) the ability to maintain high rates of operation, or large boiler capacities, by using stack draft alone or with but little assistance from mechanical draft—owing to the absence of a thick fuel bed.

(**c**) Some of the early **difficulties encountered,** which had to be overcome or minimized to make pulverized coal commercially applicable to steam generation, arose from (1) the *small combustion spaces* that failed to provide for the great length of flame produced with such fuel; (2) the rapid *deterioration of the refractory furnace walls* from exposure to the high furnace temperature and from erosion or washing away of the furnace lining due to molten fly-ash flowing over the wall surfaces when the operation was at the high rates desired; (3) the *ashpit difficulty* due to the formation of a fused mass of slag in the bottom of the pit, the mass being very difficult to remove; (4) the *large amount of fly-ash* or cinder carried by the furnace gases, some of which was deposited on the furnace walls, as already noted, some coated the boiler tubes and even bridged between them, some collected in pockets in the gas passages, and the remainder was discharged from the chimney in such quantities as to become a nuisance in certain localities. Often as much as 90 per cent of the ash is carried away from the furnace by the gases, only about 10 per cent being deposited in the furnace bottom.

These difficulties were overcome mainly by using relatively large furnaces with special air-cooled or water-cooled walls; by installing slag screens or air-cooled or water-cooled furnace bottoms, or by using slag tapping (the deposit in the ashpit being drawn off in the molten state);

by adopting suitable arrangements of boiler and flue passages; and by passing the gases through dust eliminators before discharging them.

(**d**) In addition to the consideration of the advantages and difficulties of design and operation just enumerated, attention must also be given to the **cost factors** involved. The initial expenditure for some of the equipment may be relatively large because of (*a*) the requirement of a large furnace of expensive construction; (*b*) the apparatus essential for coal preparation and handling and which is not used with other methods of coal burning; and (*c*) the extra building space required for (*a*) and (*b*). However, the initial cost can be reduced, or the excess compared with the outlay for equipment with other combustion methods may be offset, either partly or wholly, by (1) using cheaper fuel than that possible with other combustion methods;[2] (2) simplifying, reducing and combining the preparation equipment so as to decrease its cost and bulk; (3) increasing the efficiency of the steam-generating unit and the economic rate of operation, so that a smaller unit can be used for the given output; and (4) combining these methods.

(**e**) As the efficiency of the steam-generating unit is raised principally by using less excess air and high flame temperature, and as forced operation results in a more elevated flame temperature and in more fly-ash, the ultimate limitations to the output of a unit are largely fixed by size and design of the furnace and by the provision made for avoiding ash trouble. To select or to design proper equipment for burning powdered coal or to operate that equipment to best advantage requires a knowledge of the combustion process which such fuel undergoes.

422. The Combustion of Powdered Coal.—(**a**) During the combustion of powdered coal there occur numerous thermo-physical, thermo-chemical and thermo-dynamic processes which progress at various rates and partly overlap, but about which there is as yet a great dearth of scientific data. Hence, at present, there is but little intimate knowledge as to the details of just what actually is taking place while a fine particle of coal is burning. However, some progress has been made in determining the more important factors that have a direct bearing on the combustion of powdered coal in furnaces, and an attempt will be made to give a brief summary of the information now available.

In general, when a particle consisting of the mixture of heavy hydrocarbons that constitute coal enters the furnace, it is heated mainly by radiation from the flame of the fuel that is already burning. This heat first drives off the moisture and breaks the heavy hydrocarbons up into coke and lighter hydrocarbons which distill off in the form of gases and tar vapor; then, when this volatile matter, mixed with sufficient oxygen, attains a sufficient temperature, it bursts into flame, and not until this part is mostly consumed do the residue of fixed carbon, and the soot resulting from oxygen deficiency, begin to burn. The carbon in the particles remaining after the distillation of the volatile matter appears to possess characteristics quite unlike those of any other form of carbon with which we are familiar. It is of much less density than coke or coal and commonly becomes inflated into empty spheres, called "**cenospheres,**" which are light in weight and buoyant, and have a high degree of "reactivity" or avidity for oxygen.[2a]

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[2] It is interesting to note that often a 50-cent reduction in the price of coal per ton lowers the cost of generating steam more than does a 10-per cent increase in the efficiency of the unit, and is more valuable than a reduction of about $5.00 per kw. in the initial cost of the steam-generating plant.

(b) Pulverized coal floating in air, even though very explosive when in proper proportions with the air, is far from resembling a combustible gas. In a gas, the combustible is composed of individual widely spaced molecules with each of which the few molecules of oxygen needed for reaction can come into intimate juxtaposition by diffusion and homogeneous mixing; whereas, even a very minute particle of solid carbon is quite mountainous when compared with the size of oxygen molecules, and it presents but small surface to the enormous number of the latter needed for its combustion. Although some of the oxygen molecules are, of course, in close contact with the solid and can react quickly, most of them are relatively distant and must work their way through the inert enveloping gas film, consisting of nitrogen, carbon dioxide and water vapor formed by the preceding reaction; also, as the combustion proceeds, the number of active oxygen molecules left around the particle decreases, and it becomes increasingly difficult for them to make contact with the combustible remaining. Once the oxygen and combustible are brought into contact at ordinary furnace temperature the chemical reaction occurs very rapidly, i.e., the combustible can be burned as quickly as the free oxygen is brought in contact with it. Thus, the problem is to get the oxygen to the surface of the solid fuel rapidly in order to obtain high combustion rates and short lengths of flame. Fineness of pulverization and turbulence are the main expedients used to solve this problem.

(c) The **size of particle** influences both the distance that the farthest molecule of oxygen consumed must travel to make contact with the solid combustible, and also the extent of surface exposed by the particles per unit of weight. The *distance factor* is dependent on the volume of air which must be used for the combustion of the particle. Mathematically expressed: to burn a spherical particle of coal of diameter d by using air at 2000 deg. fahr., requires[3] a sphere of air having a diameter equal to about $36d$. Hence the smaller the particle the more readily does the oxygen get to it. How the relative *surface exposure* is influenced by the fineness of pulverization is shown as follows: Assuming the particle to be a sphere as before, its volume is $\pi d^3/6$, its surface is πd^2, and the ratio of surface to volume is $6/d$; hence the total surface presented by a given weight of solid combustible varies inversely as the size of its particles, i.e., the finer the pulverization the relatively more extensive is the surface. If the size of particles could be reduced to that of a molecule the conditions would be ideal from the standpoint of combustible alone. But there are both *physical and economical limits* to the fineness to which coal can or should be pulverized. No means are available by which coal can be reduced to dimensions approaching those of a molecule, and very fine particles tend to cluster together, thus partly defeating the purpose of fine pulverization; also fine pulverization is costly.

[2a] See "How Pulverized Coal Burns as shown by the Microscope," by Douglas Henderson, *Power*, July 28, 1931, p. 116.

The *size of particle* affects nearly every phase of the burning of pulverized coal. There is a definite limit to the size that will remain in suspension in the air while being burned; the size affects the *speed of ignition*, for small particles offer less time lag in the penetration of heat to the interior, and less opacity (or higher rate of radiant heat absorption) than do large ones; and the coarser the fuel the higher must be the combustion chamber temperature for a given rate of ignition (e.g., with one fuel, particles $\frac{5}{1000}$ in. in diameter required a temperature of 2600 deg. fahr. to give the same ignition rate obtained with particles $\frac{2}{1000}$ in. in diameter at 1300 deg.). The time required for the combustion of the ignited particle, and the length of flame, also depend on the fineness of the particle.[4] The fineness of coal particles to be used in a new installation has a bearing on the size and design of the pulverizing mills, burner equipment, and furnace used. The fineness also affects the cost of the coal preparation. The economic sizes which should be used with different fuels are given in Table LII in Sect. **423 (e)**.

(d) The **ignition lag** or **delay,** is the time required for the fuel to ignite after entering the furnace and is the only measure of the so-called "speed of ignition"; it is the time required to heat the "dust cloud" of entering air and coal to the ignition temperature of the fuel. In this connection, note that many experiments show that the ignition temperature of a coal is very difficult to determine, and that while comparable physical ignition points may be obtained in the laboratory under uniform conditions, yet under actual conditions ignition does not depend altogether on the fuel alone but also on a number of outside influences. The ignition lag of the dust cloud is affected by everything that influences the *rate of heat absorption* by the coal particles; these factors are: (1) the temperature within the furnace, (2) the thermal conductivity of the coal, (3) the temperature of the entering dust cloud, (4) the weight of air per unit weight of fuel, and (5) the opacity of the dust cloud.

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[3] Assuming that the coal has a H.H.V. of 13,500 B.t.u. per lb. and that 10 lb. of air are required per pound of fuel.

[4] For a résumé of theoretical studies and actual tests on the influence of the fineness of the coal, see "Pulverization and Boiler Performance," by E. H. Tenney, A.S.M.E., 1932, FSP-54-7, p. 55.

In addition to these heat-absorption factors, the ignition lag is affected by (6) the rate at which the *surface* of the coal particle undergoes *pyrogenic decomposition,* during which gases are liberated and auto-oxidation begins. This oxidation hastens the rise in temperature of these gases and also of the solid particle. Extra oxygen soon reacts with these gases at such a rate that the result is called " ignition." The exact degree of gasification that precedes ignition is not known; but the latest evidence indicates that the ignition delay is so short, 0.05 to 0.2 sec., according to Rosin,[5] that there is not time enough to permit much decomposition to precede ignition because the thermal conductivity of the coal is so low. The ignition delay may be reduced by maintaining a high furnace temperature, preheating the air, using considerable turbulence and having the coal particles of small size.

(e) **Intensive mixing** is desirable so that the comparatively heavy coal particles will have *relative motion* with respect to their enveloping gases, whereby the products of combustion and the volatile matter evolved may be swept away from the surface of the fuel and the free oxygen may be brought to the surface rapidly, thus increasing the rate of combustion. Mere turbulence alone may not be sufficient to accomplish the result desired. The mixing must be of such character as to produce the needed *scrubbing* action. Very fine particles tend to float in the ambient gases without having motion relative thereto, but, fortunately, they require less scrubbing and are more quickly consumed than are the heavier particles. The latter, owing to their inertia, tend to move with linear velocities departing from those of the enveloping gases, and if there is turbulence they move across the gas stream; further, the relative motion may also be augmented by the action of gravity. The mixing usually becomes more feeble as the distance from the burner increases, and large particles may pass out of the combustion region before they become completely consumed. When secondary air is used, it is commonly injected into the coal-carrying primary stream with a velocity of from 60 to 150 ft. per sec. and in a crosswise direction thereto. Mixing or turbulence effectively applied not only reduces the length of flame, but also causes the burning mixture to fill the furnace volume more completely than would otherwise be the case, thereby making all portions of the combustion space more effective and permitting the use of a small furnace for a given output.

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[5] " Thermodynamics of Powdered Coal Combustion," International Coal Conference Papers, 1928, p. 838. Abstracted in Power Plant Engineering, March 15, 1930, p. 326.

But turbulence is easier to initiate than to maintain, and even with intensive mixing the rate of combustion is rapid only near the burner and decreases quickly as the mixture moves away from that point, the rate becoming very low when the combustible content has been reduced to 1 per cent of its original value. It is the consumption of the last traces of combustible that requires a long time and large furnace volume. Complete combustion is not practicable or possible in commercial furnaces of permissible proportions; hence the unburnt fuel in the flue dust is never reduced to zero. The excess air has little influence on the initial combustion, but is very useful in burning the last portions of the fuel.

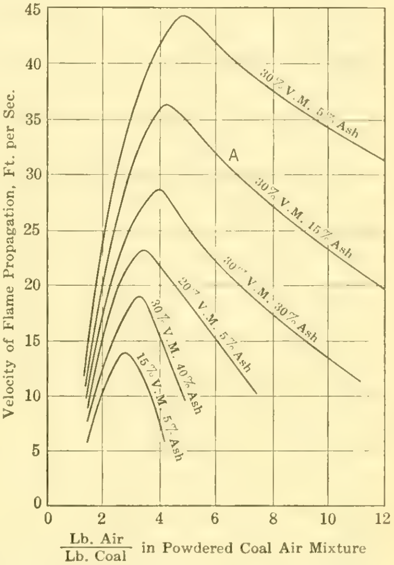

FIG. 594.—Velocity of Flame Propagation with Powdered Coal.

(f) The **velocity of flame propagation** with powdered coal of given fineness depends on the percentages of volatile matter and ash in the coal and on the weight of air supplied per pound of coal issuing from the burner. The inter-relation is indicated by the curves in Fig. 594 by DeGrey.[6] Although the chart is for a mixture with very fine coal and with combustion occurring in an unheated space, the curves may be taken as illustrative in a general way of the velocities of flame propagation within furnaces. Referring to curve *A*, for example, observe that the maximum flame velocity is about 35 ft. per sec. and occurs when a little over 4 lb. of air enter with each pound of coal dust. Thus, when this coal and proportion of air are discharged into the furnace with a velocity of 35 ft. per sec. the point of ignition will be just at the mouth of the burner. This, however, would cause destruction of the burner.

[6] See p. 45, Bul. 237, Bu. of Mines, 1925, by Kreisinger, Blizard, Augustine and Cross.

Ignition is delayed to a point farther in the furnace by using higher injection velocity, coarser particles, wetter fuel, and/or a larger air-ratio—the latter because more air must be heated to the ignition temperature. If there are variations in these conditions the flame will travel on the stream of the mixture, causing puffing or pulsation, and may even strike back into the burner. With the coal used in obtaining curve A, inflammability of the mixture is quickest when only about 40 per cent of the total air is used as *primary air* to inject the powder into the furnace, the rest being furnished as *secondary air.* With anthracite, still less air should enter with the coal. But it is not always desirable to use the maximum flame velocity and in that case a larger percentage of excess air than the optimum is, of course, required. Moisture causes particles to stick together, and, like ash, absorbs heat—hence it is supposed to retard combustion. Fineness increases the rate of propagation more in coals low in volatile, or high in ash and moisture, than in other coals.

(g) The **length of flame** of burning powdered coal—a factor which has an important bearing on the furnace size and proportions needed—depends, in general, on the percentage and composition of the volatile matter in the coal, the fineness of pulverization, the rate of combustion, the amount of air mixed with the coal, the ambient temperature and the turbulence maintained. Stream-line flames may be from 30 to 60 ft. long, whereas with turbulent combustion the length may be reduced to 10 ft. or less. The **visible flame** is produced during the combustion of the *volatile* content of the fuel, the visibility being due to the presence of small particles of incandescent carbon formed by the breaking down of heavy hydrocarbons. Visibility ends where the combustion of the volatile matter is completed or where the glow of ash particles ceases. The combustion of the fixed carbon is without visibility, i.e., with a so-called **" invisible flame."**

The distance from burner outlet to the tip of the invisible flame depends not only on the time required for combustion but also on that needed for ignition. According to Rosin,[5] the maximum burning time in furnaces of stationary boilers is from 1.5 to 3.6 sec. (In German railway locomotives it is from 0.3 to 0.4 sec.) The time for transforming the fuel into readiness for combustion depends mainly on the thermal conductivity of the fuel and requires, according to the same authority, from 0.05 to 0.2 sec.—a relatively long delay.

The period required for the volatile matter to be ejected from the particle of fuel and then to be burned is short compared with that needed for the consumption of the residue of carbon. The relative amounts of time required for combustion by the respective volatile and solid components depend on the proportionate percentages of these constituents in the fuel; and, strange as it may seem, the time of combustion appears to *increase* as the ambient temperature is raised.[7] According to the analysis by Burke and Schumann,[7] the time of combustion and length of flame with 30 per cent excess air are roughly half the values resulting with 10 per cent surplus; supplying more than 30 per cent excess does not cause much further reduction in them.

Obviously, the flame length depends on the coarsest of the fuel particles undergoing combustion, and is reduced by finer and more uniform pulverization, by having suitable ambient temperature and excess air percentage, and by employing turbulence. Long flames can be made to have helical or U-paths so that the furnace form will be appropriate for the space available in a steam-generating unit.

423. Suitable Fuels.—(a) The fuels that have been burned in powdered form are anthracite, semi-bituminous, bituminous and sub-bituminous coals, lignite, asphalt, petroleum coke, coke from high- and low-temperature carbonization, and coke braize; and these have been used both alone and in combination with gas, oil, shavings and sawdust, bagasses and other solid fuels. Fuels impossible or difficult to burn satisfactorily by other methods have been used in this form. The " fines " of the better coals, which usually sell cheapest, are often preferable, for pulverization, to the coals of more expensive size. It is to be noted, however, that not all these fuels have been burned in furnaces and plants of the same design. A furnace and its auxiliaries which are proportioned for burning low-volatile coals should not be expected to give equal results with high-volatile ones, nor will a system designed for high-fusion ash necessarily give satisfactory performance with low-fusion ash. If adequate equipment is installed to permit the use of the poorest coal that might be selected, then generally the better grades can also be burned satisfactorily when economic conditions permit; but seldom is the converse true. Coals from a given district can often be used with substantially the same performance in a furnace that is designed for the variations to be met.

Whether or not a coal is suitable for the available combustion equipment, or vice versa, depends largely on the composition and physical characteristics of the fuel, the fineness of pulverization that can be economically employed with it, and its behavior while burning. In general, the lower the percentage of volatile matter, the larger must be the combustion space; and, as high volatile coal burns readily, it does not need to be as finely pulverized as does coal low in volatile.

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[7] This surprising fact was verified experimentally—see " Rate of Burning of Individual Particles of Solid Fuel," by H. K. Griffin, J. R. Adams, and D. F. Smith, in Industrial and Engineering Chemistry, Vol. 21, No. 9, p. 808; and was also demonstrated analytically—see " Kinetics of a Type of Heterogeneous Reactions," by S. P. Burke and T. E. W. Schumann, in the same magazine, Vol. 23, p. 406, April, 1931.

(**b**) *Anthracite and coke*, being slow to ignite and burn, require large combustion space, but even then the flue dust seldom contains less than 30 per cent of carbon. The particles of such fuels retain their shape and individuality while burning. Coals having volatile content of 6 per cent or below can not be burned efficiently in a furnace that is " cold," i.e., completely water cooled, nor with high velocity turbulent burners.

Coking coals, such as the Eastern bituminous coals, soften when burning, and their particles change form and often stick together when they collide, thus forming larger particles which lengthen the time of burning and require larger space for the invisible flame. The flue dust seldom contains less than 25 per cent of combustible, and the pieces of coke in it are often larger than the original coal particles.

Free-burning coals (e.g., Illinois) do not generally become sufficiently fused for the particles to stick together, and because they are high in volatile content the residue of fixed carbon in the particle is comparatively small; hence, but relatively small combustion space is needed beyond the tip of the visible flame. Usually the combustible in the "ash" is from 5 to 10 per cent.

Western sub-bituminous coals and lignites, when heated, crack and disintegrate into smaller particles than the original ones, and since they are high in volatile content, little fixed carbon is left to burn; hence the combustion space needed is relatively small, and the combustible in the refuse may be reduced to 1 or 2 per cent.

(**c**) The *fusing temperature* of the ash is an important consideration in buying coal for burning in powdered form in existing furnaces having refractory walls; and it is one of the determining factors to consider when designing a new furnace, as will be explained in a subsequent section.

Moisture in coal interferes with the operation of pulverization and with the transportation of the powder, besides increasing the stack loss. Pre-drying is often necessary and is accomplished by methods which will be considered in Sect. **427.**

(**d**) The **grindability** of the coal has an important bearing on the capacity, wear, and energy consumption of the pulverizing mill, and on the suitability of the coal for use with given equipment. Mr. E. G. Bailey [8] gives in Table LI illustrative data on the grindability and wear characteristics of several different coals pulverized in the same mill.[9]

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[8] " Coal for Pulverizing," in Power, Sept. 23, 1930.

[9] For methods of determining the grindability indices of coal, and tests of representative coals of the United States and Canada, see " Grindability of Coal," by R. M. Hardgrove, A.S.M.E. Fuels Division Meeting, April 20–23, 1931.

The softer coals have larger grindability indices and permit greater capacity with a given mill than the hard coals. Also there is a corresponding difference in the energy required per ton of coal pulverized. The abrasion or wear of the grinding elements varies widely with the different kinds of coal and is not necessarily dependent on hardness, as it is also influenced by properties of the ash content.

In general, coals having between 16 and 28 per cent of volatile content are commonly of high grindability (index from 80 to 120). High-volatile coals are usually of hard structure that makes fine grinding difficult and costly and causes high grinder wear. The indices for such coal range from 35 to 75, but there are so many exceptions that the volatile content is not always a definite guide. Pennsylvania anthracite shows indices ranging from 20 to 48. It is important to note, however, that not all types of mills give the same grindability indices, nor do they show the same relative amounts of wear with the different coals. Also coals having the same grindability index under one loading of a mill may have widely different indices under other loadings and therefore have varying effect on the mill capacity; hence, it may be necessary to test the grindability of the various coal samples under two or more loadings.[10]

TABLE LI

GRINDABILITY AND WEAR WITH SEVERAL DIFFERENT COALS

Kind of Coal	A Penna. Bit. (Clearfield Co.)	B W. Va. and Pgh. Bit.	C Ill. Bit.	D Penna. Anth.	E W. Va. Bit. (Logan Co.)
Grindability index.....	88	65	52	44	34
Relative grinding capacity, in per cent...	100	94	88	82	75
Relative life of wearing parts, in per cent....	52	100	27	6	77

(e) As the characteristics of the coal and the fineness of grinding affect the coal-preparation energy-consumption, and as these preparation expenditures vary with the different coals, there is for each installation one or more of the available coals that will give the best results with that equipment, due consideration being given also to the heating value and cost of the fuel as well as to the efficiency obtained.

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[10] "Factors in Coal Selection," by G. B. Gould, Power, Dec. 2, 1930, p. 886.

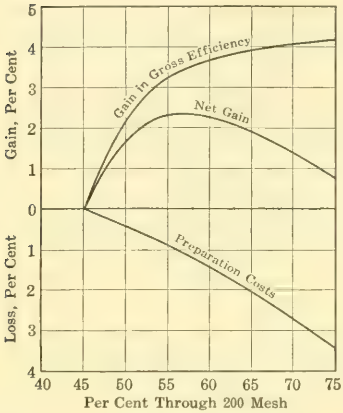

FIG. 595.—Net Gain in Efficiency of Unit vs. Degree of Pulverization.

The method of determining the best degree of pulverization[11] to use with a given coal and installation is indicated in Fig. 595. The best pulverization with other coals would be found in a similar manner; then, by comparison, the most suitable coal and fineness of grinding to use could be determined for that plant. The fineness for various kinds of coal for the best *economic* results, considering cost of pulverization as well as the losses from incomplete combustion, is indicated in Table LII, which is due to Kreisinger.[12] Such data, together with heating value and the cost of the fuel, are of value in selecting the coal to use under any given set of conditions.

(f) Fig. 596 shows the relations between screen numbers or meshes and the size of screen openings with the different *screen systems* in use.[13]

TABLE LII

ECONOMIC FINENESS OF VARIOUS COALS

	Percentages through		
	200-mesh	100-mesh	50-mesh
Anthracite	85	97	100
Eastern low volatile	75	92	98
Eastern high volatile	65	85	97
Illinois	55	80	92
Sub-bituminous	45	75	90
Lignites	35	65	88

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[11] "Pulverization and Boiler Performance," by E. H. Tenney; paper before A.S. M.E. Chicago Fuels Meeting, Feb. 10–13, 1931.

[12] "Development of Pulverized-Coal Firing and Study of Combustion;" Paper before A.S.M.E. Chicago Fuels Meeting, Feb. 10–13, 1931.

[13] From Power Plant Engineering, Nov. 1, 1930, p. 1221. A.S.T.M. = American Society for Testing Materials. I.M.M. = British Institute of Mining and Metallurgy. H & M = Howard and Morse.

Thus, " 70 per cent through a 200-mesh " means that 70 per cent of the weight of pulverized coal will pass through standard screen openings of 0.0029 in. (U. S. Std.); and a " 100/200 product" means that the particles will pass through a 100-mesh screen (openings of 0.0059 in.,—U. S. Std.) but will be retained by a 200-mesh screen. As wide variation in the results of screen tests can be obtained by different shaking methods, the tests should be made in accordance with " Method of Test for Fineness of Powdered Coal," A.S.T.M. designation D197.

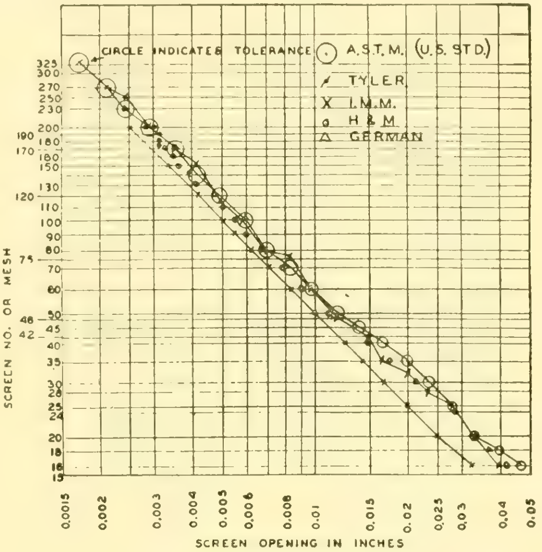


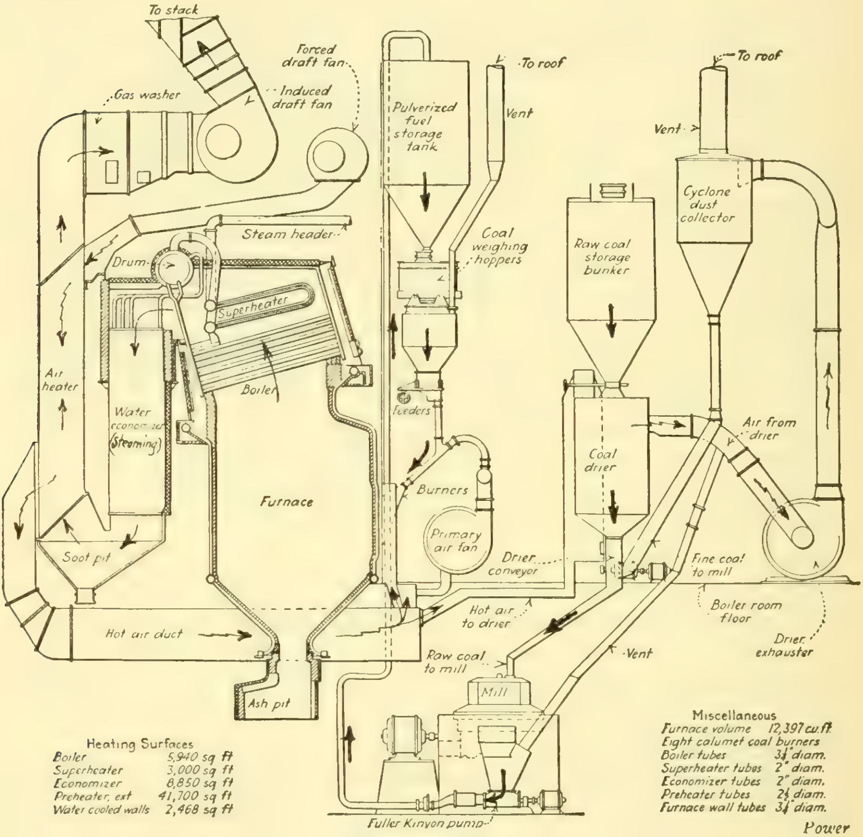
Fig. 596.—Graphical Comparison of Sieve Series.

The *shape of the particles* varies with the different coals and affects the screening results. The spherical grains have less surface than the plate-like or rod-like ones that pass through the same sieve. Also, the particles which will just pass through the 200-mesh are accompanied by others much smaller, the impalpable powder $\frac{1}{5}$ or $\frac{1}{10}$ as large being termed *superfines.* The shape of particles and percentage of superfines has an important bearing on the burning of the powder.

424. Coal Preparation and Delivery Systems.—**(a)** The primary equipment of the coal preparation plant includes (1) magnetic separators, for removing " tramp " iron which might injure the pulverizers if not eliminated, (2) crushers, (3) driers, (4) pulverizing mills, (5) storage bins for the raw and pulverized coal, (6) conveyors, for transporting the raw coal and for delivering powdered coal to the bin, (7) feeders, and (8) the burners. The supplementary equipment includes the necessary piping, separators, fans and driving motors.

There are two main systems of preparation and delivery: (*a*) the *storage system,* and (*b*) the *unit system.*

(b) In the **storage system** (also called the "bin," "indirect-firing," "central," and "multiple" system) there is a central preparation plant which delivers the powdered coal to storage bins that constitute *reservoirs* from which the coal is fed as needed to the various burners. This system uses all or most of the apparatus mentioned in the previous paragraph, although one or more of the first three items may be omitted. The apparatus can be arranged in a variety of ways. The early Milwaukee Lakeside Plant used this system, the diagrammatic arrangement of which has already been shown in Fig. 593. Other arrangements are illustrated in Figs. 539, 597 and 598. In the plant shown in the latter figure the coal unloading and the coal preparation each take place in a building separate from the rest of the plant, as shown at (*a*) in the figure.



Power

FIG. 597.—Bin and Feeder System at Calumet Station.
(See Fig. 545 for Unit System in Same Plant.)

With this system pulverizing can be done at a constant rate and at off-peak hours, excellent operating conditions may be maintained even under varying loads, and temporary mill failure will not cause an immediate shut-down; but the cost of the preparation plant and storage bins, and also of the space occupied by them, is relatively large. The energy used for operating the coal preparation equipment ranges from 15 to 30 kw-hr. per ton, and the total cost of coal preparation, including maintenance and fixed charges, is from 15 to 65 cents per ton. For other cost data see Sect. **432.**

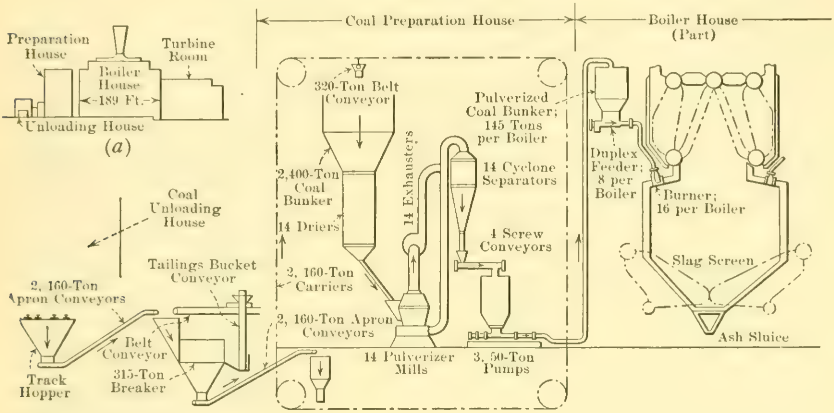


FIG. 598.—Coal Preparation System at Trenton Channel Plant of the Detroit Edison Co.

(c) In the **unit system,** the raw coal is delivered to a single piece of apparatus which contains the tramp-iron separator, feeder, pulverizing elements and fan, and from which the air-borne powdered fuel is piped direct to one or more burners, without any provision for storage. With this system it is generally not necessary to dry the coal initially, and often preliminary crushing can be dispensed with. The apparatus is relatively small and inexpensive, occupies but little space, and can be conveniently located near the furnace. A single unit mill may be used for one or two boilers if they are small and the load is fairly steady, but with such arrangement the failure of the mill will cause a shut-down. Single mills may be somewhat inflexible of control under reduced output and variable demand. With a large boiler, two or more pulverizers may be used, and the larger the number the more flexible and certain can be the operation. The energy used in driving the mills and their fans in machines ranging in capacity from 2000 to 4000 lb. per hr. is from 17 to

25 kw-hr. per ton with bituminous coals and lignites, but anthracite requires more.[14]

One arrangement of the unit system has already been shown in Fig. 545; others are illustrated in Figs. 599 and 600.

(d) Compared with the storage system, the unit system with several mills has about the same flexibility and gives practically the same efficiency under steady loads, but is unable to function as well under low loading; also the fineness of pulverization is more variable, changing with moisture, mill load, and rate of air flow through the mill; and further, the air-to-coal ratio may be more difficult to regulate.

Fig. 546 (*b*) gives efficiency curves obtained when burning one kind of

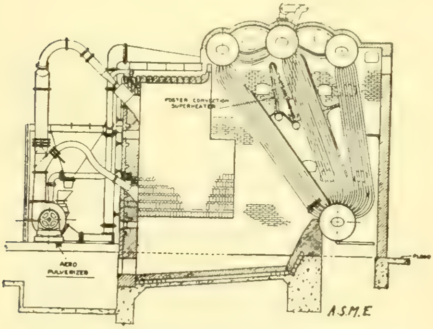


Fig. 599.—Unit Pulverizer Installation, 4000 Sq. Ft. Boiler.

(Two burners carry the normal load, and an auxiliary burner carries the low loads on Sundays.)

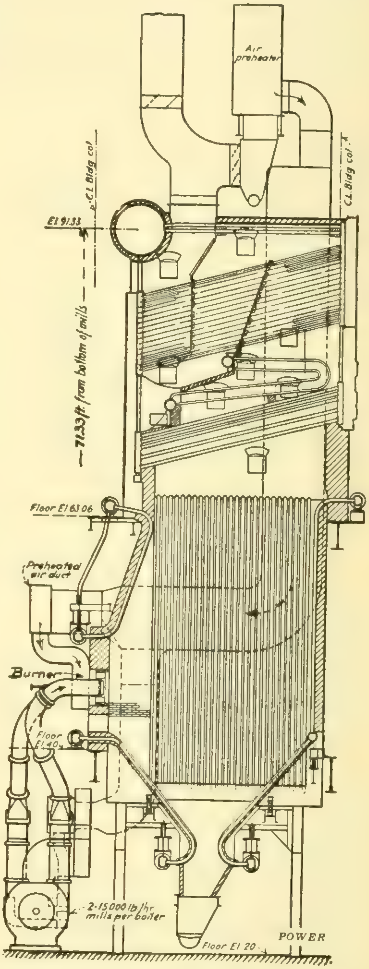


Fig. 600.—Unit Mills Used with 18,000 Sq. Ft. Boiler, Cahokia Station, St. Louis.

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[14] For an exhaustive examination of the unit system, furnaces, costs, etc., see "Pulverized Coal in Stationary Railway Power Plants," as reported by The International Railway Fuel Assoc. and published as a serial in Power, beginning June 5, 1928, pp. 1032, 1072, 1118 and 1160; also in Combustion, beginning Aug., 1928. Another reference is "Unit System of Coal Pulverizers for the Generation of Steam," by J. Blizard, Trans. A.S.M.E., 1928, FSP-50-61, p. 97.

coal with both systems in a unit which is otherwise the same, the arrangements of apparatus in the two cases being those shown in Figs. 597 and 545, respectively.[15] The difference between the two efficiency curves is quite small, hardly more than would be accounted for by errors in making observations, and might not justify the added expense for the bins and extra equipment required with the storage system; also the forms and positions of the curves might be altered somewhat by improving the design of apparatus as compared with the experimental equipment used, which was not necessarily the best.

(e) **Combinations of the two systems** may be desirable when advantage can be taken of the low first cost, simplicity, and small space required by a unit system of size suitable to meet the larger and steadier loads, supplemented by a storage system of capacity relatively small but sufficient to aid in meeting the maximum and minimum demands, to make the rate of pulverization nearly uniform, to reduce the total mill capacity required, and to provide flexibility and certainty in operation. The two systems are used in the high-pressure Deepwater (N.J.) Plant.[16]

425. Burners and Feeders.—(a) A burner should (1) mix the coal uniformly throughout the primary air, (2) cause or aid the streams of secondary air and the primary air and fuel, as they enter the furnace, to impinge on each other in such manner as to effect uniform mixing and turbulence, (3) avoid producing excessive flame velocity and length, (4) give good combustion at the minimum and maximum loads, as well as under normal loading, (5) have suitable means for regulating the fuel and air, (6) offer little resistance to the flow of material passing through, (7) expose a minimum amount of metal to furnace radiation, (8) be convenient to light, and (9) be non-clogging, or easily cleaned.

The design and operation of a burner depends on (*a*) the character of the fuel (i.e., the fineness, surface moisture, and volatile content of the coal, and the porosity and size of the coke particles formed during combustion), (*b*) the method and place of mixing the coal and the primary air, and the path that the mixture takes to the burner, (*c*) the percentage of the primary air used, (*d*) the temperature and pressure of the primary and secondary air, (*e*) the character of the load variations, and (*f*) the design of the furnace. The better the combustion conditions provided by the burner, the smaller need be the furnace volume.

The range in capacity ratio of most burners is usually limited to from 2 : 1 to $3\frac{1}{2}$: 1. Further change in generating capacity is obtained by shutting off burners in central firing, or by shutting down mills and their burners in the unit system—assuming that there are several burners or mills serving a unit.

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[15] "Comparative Performances of a Pulverized Coal-Fired Boiler Using Bin System and Unit System of Firing," by A. E. Grunert, Trans. A.S.M.E., FSP–53–4, p. 35.

[16] "Operating Experiences, Deepwater Station," by K. M. Irwin; paper before the December, 1930, meeting of the A.S.M.E.

(**b**) The velocity of the mixture of the fuel and primary air through the piping and burner must always be greater than that of the flame propagation, so as to prevent flarebacks and damage to the burner. Generally, the surface or skin of a jet issuing from an open tubular burner moves more slowly than the core because of friction with the ambient gases in the furnace; therefore, the surface ignites and burns closer to the burner than does the core, and the average velocity of the jet must be greater than the speed of the flame propagation. With such a burner the tip of the flame at the center of the core is much farther from the burner than are the boundaries of complete combustion at the sides of the flame, the flame is long and the principal isotherms are heart shaped. Turbulence tends to equalize the conditions across the stream, flatten the boundaries of complete combustion and the isotherms, and shorten and widen the flame. But to produce turbulence initially is easier than to maintain it.

(**c**) Although the ratio of primary air to coal should be kept constant at its best value, actually with *unit* firing this ratio varies over a considerable range with the load. With this kind of firing, an adjustment is generally provided at the burner outlet to secure proper velocity of the fuel-and-air mixture as it leaves the nozzle. With the *storage* system the ratio of coal to primary air is under direct control of the operator, and less adjustment at the fuel nozzle is required. Keeping the ratio of primary air to coal constant at the value giving the best flame propagation may, under heavy loading, cause the flame to be unstable or impossible of maintenance; hence, with a single burner, a smaller ratio is commonly used with heavy loads than with light ones. When several burners are used the ratio can be kept constant and the load be changed by varying the number of burners in operation. When the furnace has refractory walls, the excess air percentage must sometimes be increased under heavy loads to limit the furnace temperature. The percentage of primary air used varies with burner design, from a small value to the total supply.

(**d**) There are two **main types of burners** for use with pulverized coal: (1) the *stream-line type,* and (2) the *turbulent type.*

Stream-line burners discharge the mixture of primary air and coal with little or no turbulence and produce long flames. They are often of the *fantail* type, a simple form of which is illustrated in Fig. 601. An application is given in Fig. 602, which shows the old Lakeside Furnace.

(See also Fig. 593.) They are arranged to project the primary air and fuel downward in thin flat streams, parallel and edgewise to the front wall of the furnace. This wall is hollow and, through adjustable openings, discharges heated secondary air horizontally into the flame, thus producing mixing and turbulence, and also protecting its refractory lining. Burners of this type sometimes have ribbed, serrated, or riffled nozzles; and deflectors may be added around the outlet to cause high-velocity jets of cold or heated **tertiary air** to impinge on the primary stream. These expedients shorten the flame. The simplest form of stream-line burner for low rates of combustion consists merely of a plain cylindrical tube with round or flattened orifice. Generally, the velocity of the primary stream is from 50 to 150 ft. per sec., and the velocity of the secondary is from 20 ft. per sec. with induced draft, to over 100 ft. per sec. with forced injection of heated air.

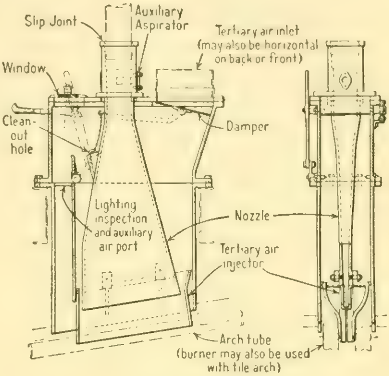
Slip Joint
Auxiliary Aspirator
Tertiary air inlet (may also be horizontal on back or front)
Window
Clean-out hole
Damper
Lighting inspection and auxiliary air port
Nozzle
Tertiary air injector
Arch tube (burner may also be used with tile arch)

Fig. 601.—Vertical Fantail Burner. (Lopulco, C. E. Co.)

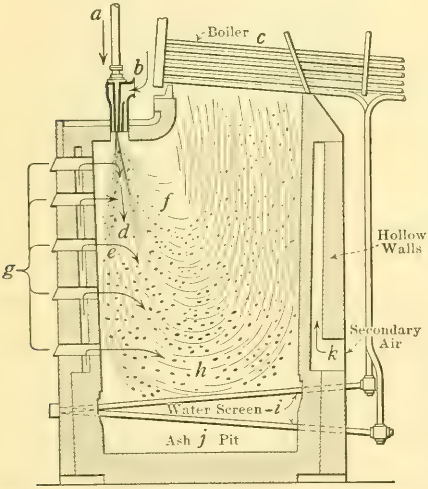

Fig. 602.—Fantail Burner and Furnace (Lakeside).

a, Coal and 10 per cent air; *b*, 20 per cent air; *c*, boiler, first pass.; *d*, velocity 100 to 150 ft. per sec.; *e*, velocity 10 to 15 ft. per sec.; *f*, fine coal particles; *g*, about 65 per cent air; *h*, coarse coal particles; *i*, water screen; *j*, ash; *k*, air to hollow walls.

(e) In **turbulent burners,** both the coal-bearing primary air and the secondary air pass through the burner, and they are thereby made to mix intimately as they enter the furnace together, thus giving a flame that is relatively short and flaring compared with that from a stream-line burner. There are many forms of turbulent burners, some with circular discharge orifices and others with straight narrow outlets. The secondary air is drawn through some of them by induced draft; in others it is forced through under pressure. **Circular burners,** one simple form of which is shown in Fig. 603, commonly have a central pipe or nozzle for the primary air and coal, and some kind of a diffuser at its outlet; around this central portion is a windbox which delivers the secondary air through a concentric annular opening, which frequently contains deflectors or vanes. Means for regulating the flow of fuel and air are always provided, and the deflectors are often made adjustable. These burners, which can be of large capacity, are usually arranged to discharge horizontally and through refractory walls, as in Fig. 600. Fig. 604 shows a burner of this type that can use either powdered coal, oil, or gas, as the fuel. In this burner the coal-air mixture issues annularly around a central core of secondary air.

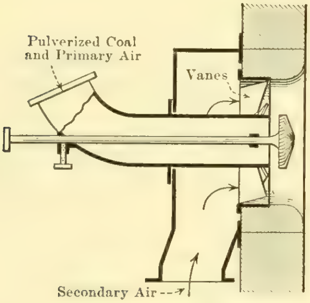

Fig. 603.—One Form of Circular Burner.

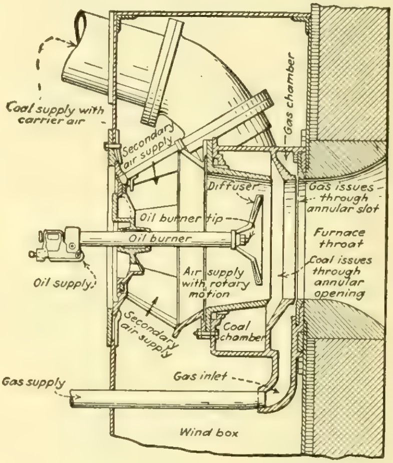

Fig. 604.—Burner for Powdered Coal, Oil or Gas.

A **turbulent intertube burner** designed to discharge between the tubes of water-cooled furnace walls is shown in Fig. 605. From it the primary air with its coal issues in a long, narrow, inclined stream which is broken up by jets of secondary air discharged under pressure through oblique ports arranged alternately on either side. Refractory faced plates separate the air ports. Applications of burners of this type are shown in Figs. 545 and 597. In general, this type of burner is used in furnaces which have completely water-cooled walls and bottoms and which are designed for coals having ash of high fusing temperature. The **cross-tube burner** of the turbulent fantail type, shown in Fig. 606, has a horizontal slot which discharges the primary air and fuel through the spaces between the vertical water tubes of the furnace walls, either horizontally or at a downward angle. The secondary air enters the furnace through adjustable port openings above and below the primary ports in such manner as to cut the primary stream and distribute the flame across the width and length of the furnace. This burner was originally designed for furnaces with slag-tap bottoms, in which the slag must be kept in a molten condition.

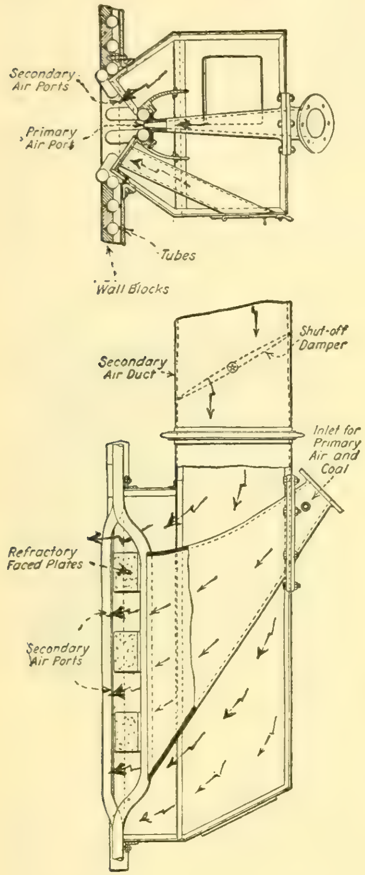

Fig. 605.—Calumet Intertube Burner.

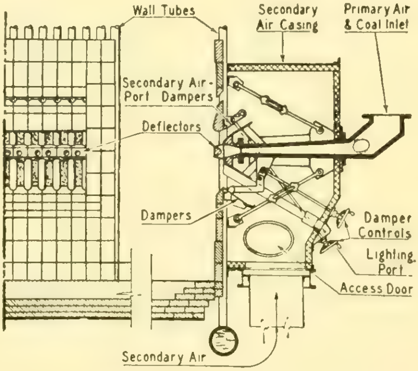

Fig. 606.—Cross-Tube Burner (Fuller Lehigh).

(f) **Feeders** for introducing the powdered coal into the primary air at regulated rates are necessary with the storage system, whereas with the unit system they are not needed because the carrier-air, which sweeps the powdered coal out of the unit mill as fast as it reaches the required fineness, conveys the powder direct to the burner. As dry powdered coal is very fluid and will readily flow out of even very small holes or cracks, the feeder must be designed to prevent flooding or running.

In one arrangement, Fig. 607, the feeding is done by means of a long, motor-driven screw at the end of which is a spring-loaded disc, serving as a " regulator" to prevent the coal from running through the feeder and flooding the system. The primary air, under a pressure of from 8 to 20 in. of water, sweeps past the end of the screw, picks up the coal and carries it through the discharge pipe which leads to the burner. Dampers are provided for controlling the primary air delivered to the burner. In another arrangement of feeder, Fig. 608, a vertical shaft drives two star wheels which receive coal at one place in their revolution and carry it around on plates to the discharge opening at another place, thus preventing flooding. The coal from the second star wheel enters the burner pipe, where it is entrained by the primary air. Both of these feeders are attached to the bottom of the pulverized-coal hopper, have louvre gates between (so that the coal supply can be shut off when the feeder is out of service), and have agitators or fluffers which keep the coal from bridging or packing. Other feeders make use of belt or chain conveyors, revolving plates, rocking gates, revolving pockets, or reciprocating pushers. Several feeders may be built in a group operated by a single motor, with clutches between the drive mechanism and the individual feeders.

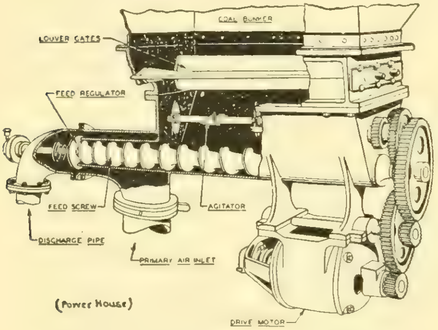

FIG. 607.—Lopulco Screw Feeder.

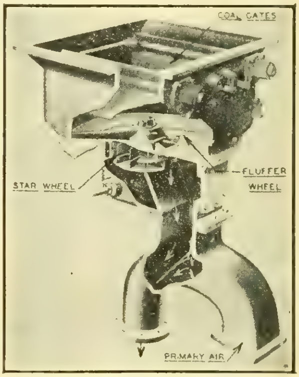

FIG. 608.—Bailey Feeder.

426. Firing Methods and Furnace Requirements.—(a) There is an inseparable relation between the burner and furnace arrangement of powdered-coal installations, for the size, shape and wall construction of the furnace must be selected with respect to the burning equipment, with due consideration given to the elements of time of burning, temperature and turbulence. By using fine coal, suitable temperature and intense turbulence the size of furnace can be reduced and the rate of heat liberation per cubic foot of furnace volume increased; but with the more intensive operation, the furnace walls will be destroyed rapidly unless they are of suitable design.

The "*method*" of firing can be with vertical, horizontal, oblique, crosswise or tangential flames, or with combinations thereof. With stream-line burners, the flame, since it is very long, is commonly given a U-path or a helical one, so that the combustion space can be of suitable shape with respect to the rest of the unit, and can be as fully used as possible. With turbulent burners the firing is usually horizontal.

(b) One arrangement with **vertical firing** has been shown in Figs. 593 and 602 (Lakeside). This is still one of the most widely used methods. Fig. 609 shows an application to a very large unit (East River) on each side of which there are vertical burners each provided with a small horizontal burner discharging into the main stream.[17] This arrangement causes quick ignition, pronounced turbulence, high temperature close to the burner, and a relatively short, intense flame which requires that the furnace walls and arches be fully water cooled, but permits using a comparatively small furnace volume. The secondary air is injected under pressure around the vertical burners and through the furnace walls. Such an arrangement is particularly suitable with low-volatile coals.

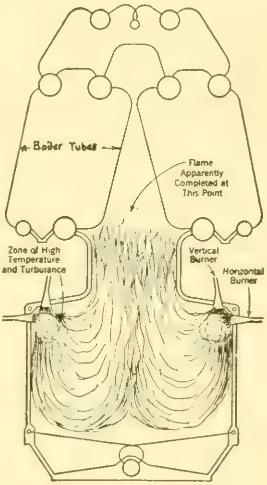
Boiler Tubes
Flame Apparently Completed at This Point
Zone of High Temperature and Turbulance
Vertical Burner
Horizontal Burner

Fig. 609.—Combination of Vertical and Horizontal Firing.

(East River Unit—N. Y. Edison Co.)

(c) Horizontal firing with turbulent burners discharging through the side of the furnace has been shown in the preceding illustrations (Figs. 539 and 600). In large furnaces horizontal **opposed firing** is sometimes used, the burners being placed in opposite walls. This method produces a beneficial secondary mixing and turbulence. Such an arrangement is used in the large Hell Gate steam-generating unit (shown in the Boiler Chapter) which has a slag-tap bottom, and uses burners of the cross-tube type. Horizontal firing does not require having the somewhat expensive arches that are necessary with vertical firing, and the burners used admit the secondary air as well as the primary mixture. The installation of large circular burners is simplified if the front wall is made of refractory material, as in Fig. 600.

(d) Tangential firing, Fig. 610, is that system in which one or more burners are located in each of the four corners of the furnace and are arranged to discharge horizontally in a direction tangential to an imaginary circle at the center of the furnace. With this method of firing the flame stream from one corner impinges on that from the adjacent corner, thus producing intense turbulence and a flame travel that is almost cyclonic in appearance. The flame is short, and completely fills the furnace, and the heat liberation is very high per unit volume of combustion space. The combustion is complete by the time the whirling mass has ascended about 3 ft. above the level of the burners. The burners can be of the simplest form, the secondary air being injected under pressure around the primary nozzle and in the same direction as the primary stream. The furnace walls must, of course, be completely water cooled. If the burners are placed near the bottom of the furnace and the mixture moves upward the heavier particles of coal which are thrown outward against the furnace walls are kept from falling by the helical ascent of the gases. Tangential firing is used in the Combustion steam generator shown in the Boiler Chapter.

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[17] "Pulverized Fuel Firing Methods," by G. W. Clendon, in Combustion, May, 1931, p. 36.

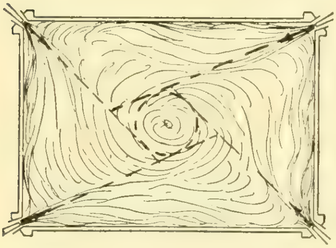
Fig. 610.—Tangential Firing.

(e) **Bottom firing** has been applied in a few installations; it simplifies the coal piping, as but a single large central burner is used. The coal and air are given a rotary motion as they issue from the burner, and hence completely fill the furnace, but there is no impingement of streams to produce mixing. Turbulence can be caused by injecting streams of secondary air through the corners of the furnace, as in the tangential method of firing.

(f) The furnace, method of firing, type of burner, character of fuel and load conditions are interdependent. The volume of **combustion space** required depends both upon the rate at which the heat must be liberated within the furnace, and upon the rate per cubic foot of furnace volume that the furnace walls and other parts will withstand. According to Kreisinger,[18] at present, good engineering practice is to design *refractory* furnaces for rates of heat liberation not exceeding 20,000 B.t.u. per hr. per cu. ft. for continuous operation; and with fusible ash, the rates generally do not exceed 15,000. With *water-cooled* furnaces the rates are limited to 30,000 for continuous operation, with a peak rate of 35,000. Higher rates are the exceptions and require extra good coal and fine pulverization. Kreisinger also states that:

> "The limit to the rate of heat liberation in refractory furnaces is the excessive erosion of the refractories by molten ash resulting in high furnace maintenance and difficulty in removing the refuse from the furnace. In water-cooled furnaces the limit is generally the deposition of slag on boiler tubes and the clogging of the gas passages through the boiler. Losses from incomplete combustion in many cases are not a serious limitation. With intensive mixing coal could be burned at considerably higher rates before the economic limits due to incomplete combustion would be reached."

[18] "Development of Pulverized Coal Firing and Study of Combustion," paper before the A.S.M.E. Fuels Division, Feb. 10–13, 1931.

(g) The **type of furnace construction** ranges, according to the condition imposed, from that having walls wholly of refractory material, perhaps air cooled, to the fully water-cooled construction, often with facing of metal or refractory blocks; and frequently part of the walls are refractory and the rest are water cooled. As the water tube walls become covered with an incandescent coating of pulverized ash when the operation is intensive, there is less need than was formerly supposed to install refractory surfaces for promoting the combustion of low volatile fuels when they are to be burned at high rates. This need is small also because the magnitude of the radiation from furnace walls changes but little with large variations in the temperature of the wall surfaces (even with a change as much as from 230 to 2012 deg. fahr., as shown by experiments with gas)[19] and because the influence of the walls extends only about 6 in. from the centers of the tubes,[20] owing to the opacity of the flame mass. Furnace construction will be considered in greater detail in a subsequent chapter.

427. Drying.—(a) Surface moisture in coal which is to be pulverized is objectionable, especially if present in considerable proportion, because it (1) decreases the efficiency of the steam generating unit, (2) lowers the capacity of the pulverizing mill, (3) increases the energy expended in grinding, (4) makes difficult the transporting of the powdered fuel; and because it may also cause (5) the coal to pack and arch in the storage bin, (6) the particles to stick together (" snowball ") in certain types of feeders and burners (hence defeating the purpose of fine grinding), (7) the ignition and flame propagation to be retarded, and (8) corrosion of bins and conveying apparatus to be increased, which adds to the cost of maintenance. Surface moisture is more detrimental with the bin system than with the unit system. Early installations, especially of the bin system, almost always included separate apparatus for drying the raw coal before pulverizing it, but now the coal is quite commonly dried in the mill while it is being pulverized.

(b) **Independent drying** may be accomplished in the following ways: (1) by applying to the coal sufficient heat from an external source to cause the water within the lumps to vaporize, force itself to the surface and pass out as steam; (2) by causing heated gases to flow through the coal to evaporate and carry away the moisture; and (3) by a combination of these methods.

Using fuel to dry the coal by the first method alone is not recommended in general. Steam is used in some driers in which the coal descends by gravity over hollow grids through which low-pressure steam passes, and air is drawn crosswise through the coal to carry off the moisture. In other steam-heated driers the coal is moved mechanically, thus avoiding the difficulties encountered in maintaining the flow through the gravity type of drier. In one arrangement using the second of the three methods the coal descends by gravity in thin sheets through a vertical stationary drier, while hot gases from the steam-generating unit are passed across the layers, provision being made for using steam for extinguishing any fires that may start. The third, or combination, method was applied in the early Lakeside installation, shown in Fig. 593, in which the drier is of the kiln type consisting of an externally fired revolving drum, slightly inclined from horizontal, through which the coal and the gases from the heating furnace pass in opposite directions, the gases carrying away the moisture. There are many other possible arrangements of these types of independent driers, but all are rather costly and occupy much space, their lengths ranging, for example, from 20 to 40 ft. In modern practice, mill drying is often substituted.[21]

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[19] Rembert and Haslam, in Industrial and Engineering Chemistry, December, 1925.

[20] " Temperatures in Powdered-Coal Furnaces Having Extended Radiant Heat-Absorbing Surfaces," Progress Report of A.S.M.E. Committee on Boiler-Furnace Refractories, by R. A. Sherman, Mech. Eng., Vol. 49, 1927, p. 335.

(c) In **mill drying**, the coal, while being pulverized, has its moisture removed by passing cold air, hot air or flue gas through the mill, the air or gas also acting as the carrier medium for removing the powder as fast as it becomes fine enough. A small amount of drying can be effected even with unheated air. In the unit system the carrier air transports the fuel direct to the burner and is used in the combustion. With the storage system this medium usually carries the powder to a separator, commonly of the cyclone type, in which the fuel is deposited. The gases are often recirculated, as in Fig. 611, when only a small percentage of moisture is to be removed; but then a certain amount of venting is necessary, otherwise the gases become saturated

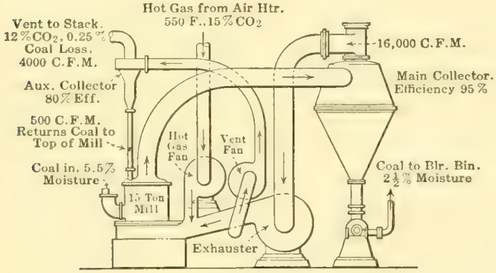

Fig. 611.—Flue-Gas Mill Drying Installation. (1929 Lakeside Extension—N.E.L.A. Report)

and ineffective for removing further moisture. Also sweating and corrosion may take place, especially if heated air or gas is used. The amount of bleeding needed in each case depends on the initial moisture in the coal and on the relative humidity and temperature of the air or gas. The discharge of coal dust from vents constitutes a nuisance if not properly cared for. Vented air can be used as primary or secondary air in the burner.

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[21] For typical designs and performance curves of the various types of coal driers, see "Better Boiler Room Operation," by Otto deLorenzi, in Power House, November 20, 1929; also see "Eliminate Moisture in Pulverized Coal," in Power Plant Engineering, January 1, 1929, p. 48, and the N.E.L.A. Reports on Pulverized Fuel.

The amount of drying necessary depends on such factors as the initial wetness and other characteristics of the coal, the type of mill used, and the fineness of pulverization desired. Unnecessary drying is wasteful.[22] The effect of moisture and air-bleeding is illustrated in a general way in Fig. 612, which applies to one kind of mill used with one kind of coal.[23]

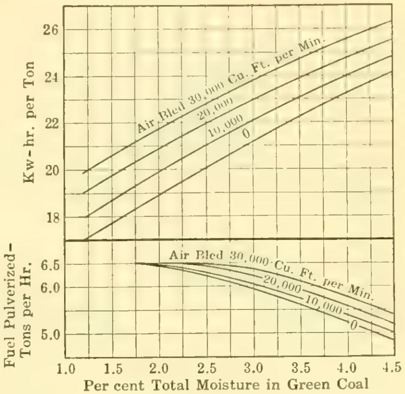

FIG. 612.—Effect of Moisture and Air Bleeding on Performance of One Kind of Mill with One Kind of Coal.

428. Pulverization.—(a) As the fineness of pulverization affects nearly all phases of the burning of powdered coal, the design and performance characteristics of the *pulverizing mill* (also called *pulverizer* or *mill*) should receive careful consideration. A mill should operate over a long period of time before replacements of worn parts become necessary in order to avoid having unsatisfactory fineness of pulverization or mechanical trouble; its design should facilitate making repairs or substitution of parts when required; the energy consumption should be reasonably low over the whole range of operating conditions; and the cost of maintenance should be small.

(b) Usually the green coal is freed of " tramp iron " (Fig. 613) and is crushed between rolls (Figs. 613 and 614) before delivery to the mill; and it is dried if necessary. The fineness of the powder delivered by a mill is controlled by the rate of coal feeding, by using screens or classifiers, by air separation, or by combinations of these methods. In air separation a stream of air sweeps through the mill, gathers up the dust, and passes through a separator which removes the coarse dust and returns it to the mill. The screening or separation may be done within the mill or external thereto, and the fan for circulating the carrier air may be incorporated as part of the mill or may be separate from it. **Unit mills,** designed for use with the unit system, especially in the smaller sizes, may have the complete preparation equipment entirely within them, i.e., the mill housing may contain the tramp-iron separator, feeder, crusher,

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[22] For method and chart to use in computing the air requirement with mill drying, see " Mill Drying of Pulverized Coal," by B. J. Cross, in Combustion, June, 1931, p. 48.

[23] Proc. N.E.L.A., 1926, p. 1432.

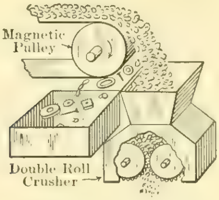

Fig. 613.—Magnetic Separator.

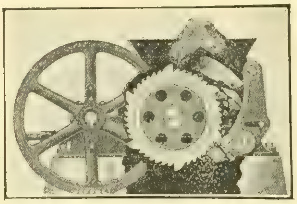

Fig. 614.—Single-Roll Coal Crusher.

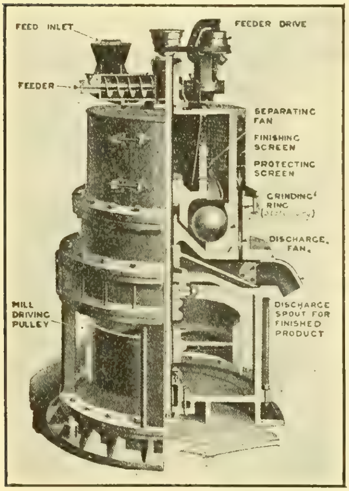

Fig. 615.—Small Fuller-Lehigh Mill with Rolling Balls, and with Air and Screen Separation. Large mills have upper and lower rows of grinding balls.

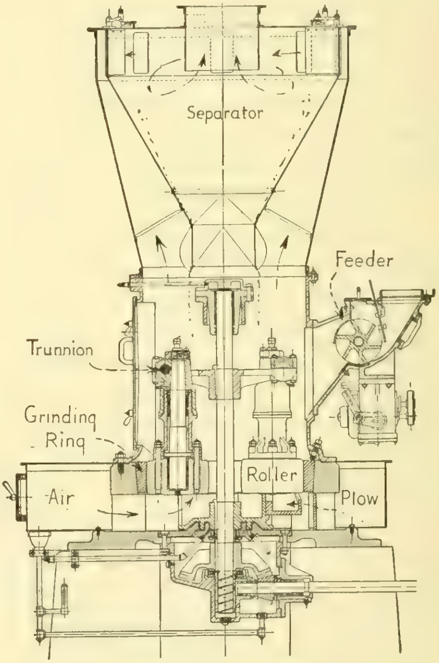

Fig. 616.—Raymond Roller Mill.

pulverizer, rejector and fan, the latter forcing the carrier air through the apparatus and delivering the fuel-and-air mixture direct to the burner.

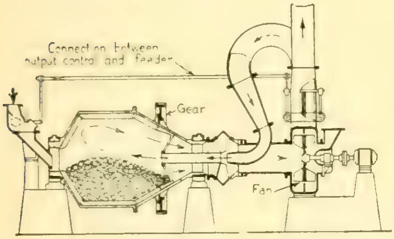

Fig. 617.—Hardinge Ball Mill (Tube Mill).

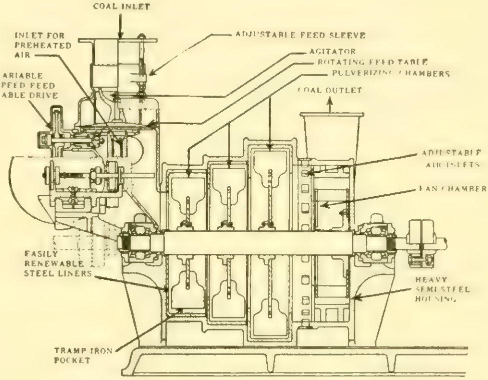

Fig. 618.—" Unitype " Pulverizer (Erie City).

(c) **Pulverizers** or **mills** reduce the coal to powder by using impact, crushing, grinding, attrition, or a combination of these. The mills must be of rugged construction and are generally driven by electric motors. They usually contain (1) revolving swinging-hammers or rings, (2) revolving rigid arms, paddles or pins, (3) rollers or balls that roll on or inside of stationary pulverizing rings, (4) balls or rods that are dropped or cascaded with the coal in a revolving drum or tube (as in " tube mills ") or (5) grinding surfaces which move relative to each other. There are a great many different arrangements of pulverizing mills, the elements of only a few of which can be shown here. See Figs. 615 to 619, inclusive. There are many manufacturers of pulverizers, some of whom make several types, designs and sizes to suit the various conditions encountered. A detailed discussion of the various makes is beyond the scope of this text.

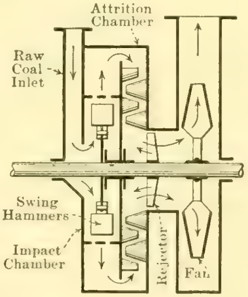

Fig. 619.—Riley "Atrita" (Main elements only).

(d) The size of mill, power required, and amount of wear for a given output are dependent mainly on the inherent physical characteristics of the coal (such as the friability, volatile percentage and the hardness), the amount of moisture (or extent of drying used), and the fineness of grinding required. The amount of drying and fineness of pulverization should be reduced as much as is consistent with obtaining satisfactory results.

It is wasteful to grind to a greater fineness than is required by the burner and furnace design and for the given operating conditions. One should remember, however, when planning a new plant, that, the finer the particles of fuel, the shorter will be the flame and the smaller the furnace. Under each set of conditions a certain balance must be found between the combustion efficiency, investment for equipment, and cost of operation, that will guide the decision as to the most economical fuel to use.

(e) The conditions under which various mills operate are widely diversified, and not many coordinated data on performances are available at present. Fig. 620 shows in a general way the wide variation in the energy used per ton of coal pulverized to various degrees of fineness by commercial machines of different types when operated at rated capacities with coals of various kinds and conditions.[24] According to *Rettinger's Law*, which is generally accepted as being approximately correct, the work done in grinding coal or other substances is proportional to the new surface produced, and hence is proportional to the reciprocal of the diameter of the particles or of the screen mesh opening.[25] The effect of grindability of the coal on mill capacity is shown in Fig. 621, due to Hardgrove.[25] Curve *A* is for a closed circuit system with coarse grinding; curve *B* is for a tube mill with external classifier and with large recir-

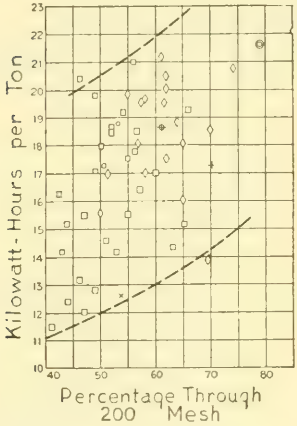

Fig. 620.—Mill Energy vs. Fineness.

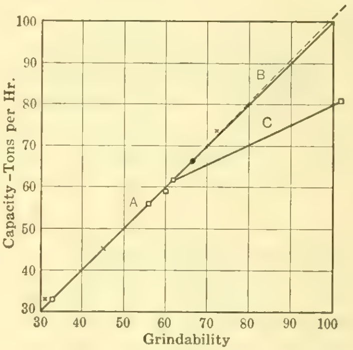

Fig. 621.—Mill Capacity vs. Grindability.

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[24] Modified from "Coal Pulverizers," by W. J. A. Lincoln, Trans. A.S.M.E., 1929, FSP-51-28, p. 159, and based on data contained in Proc. N.E.L.A., 1927. Each symbol in the figure represents a different type of mill.

[25] See Footnote No. 9.

culating load; and curve C is for an air-swept mill with internal classifier. The effect of moisture and air bleeding on mill capacity and energy consumption per ton of coal pulverized was illustrated in Fig. 612 for one kind of mill and coal.

429. Storage and Transportation of Powdered Coal.—(a) For the **storage of powdered coal,** the bins used should have sides inclined at an angle of not over 30 deg. from the vertical so as to prevent arching of the coal. When the storage period is short the material remains fluffy and free-running, but with long periods the coal packs down. The fluffiness may also be destroyed by moisture. The mass of dust in a bin easily ignites spontaneously, especially if moisture is present; but since the bins are always covered, although vented, the oxygen present does not support combustion long and the results are not serious, except that the coal may cake and lose its free-running property.

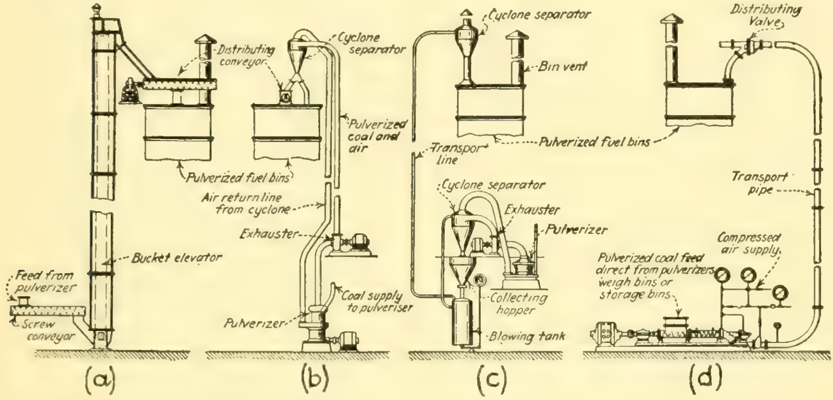

FIG. 622.—Some Systems for Conveying Powdered Coal.

(b) The kind of **transporting system** used for conveying powdered coal depends on the distance to be traversed and on the relative locations of the receiving and delivery points of the system. Powdered coal has the characteristics of a fluid and must be handled as such. Several conveying systems are illustrated in Fig. 622.

(1) Dust-tight *screw conveyors*, Fig. 622 (*a*), are often used for transporting over distances up to 200 ft. or more when there is no change of direction or elevation. This system is simple, rugged, and low in first cost and in energy consumption. Bucket elevators are sometimes used for conveying the coal from one level to another, as in the same figure.

(2) In the air *recirculation system*, Fig. 622 (*b*), as already noted, a fan produces a current of carrier air which collects the dust in the mill, con-

veys it to a cyclone separator, which removes the fuel and delivers it to a bunker below, and the air is returned back to the mill to start the cycle again. This system is limited to comparatively short distances, and the pipes required are of large size. If the exhaust fan is placed in the pipe returning from the separator to the mill, as in Fig. 611, the fan requires less energy to drive it, the blade wear is less and the leakage of dust from the system is smaller than when the fan is located as shown in Fig. 622 (*b*).

(3) In the *blowing tank system*, or *compressed-air system*, Fig. 622 (*c*), the tank is first filled with coal from a storage bin; then, with coal-supply valve shut, compressed air at pressure from 10 to 60 lb. per sq. in. is admitted to the tank and forces the coal through the delivery pipe which extends to within a few inches of the tank bottom. The tank may be mounted on weighing scales. Generally, from 700 to 1000 cu. ft. of air are required per ton of coal, which is equivalent to about 1 per cent of the combustion air; and the energy used is from 2 to 4 hp-hr. per ton. This system is intermittent and requires considerable attention.

(4) In the *pumping system*, Fig. 622 (*d*), the powdered coal is forced through a motor-driven screw or rotary pump which delivers it into small jets of compressed air (pressure from 5 to 90 lb. per sq. in.) which aerate the mass, render it fluid, and transport it readily over long distances (e.g., up to a mile), the energy requirement being from $\frac{3}{4}$ to 3 hp-hr. per ton. The delivery valves on the various bins served can be remote-controlled from a central point, where an automatic signal arrangement gives the operator information on which to base the regulation of the system.

430. Hazards with Powdered Coal.—Danger of fires or explosions exists with powdered fuel, as with any other kind of combustible dust, unless proper precautionary measures are taken. The primary factors affecting inflammability or explosibility of coal dust are: (1) the composition of the coal—there being little danger of explosion when the volatile content is below 10 per cent, as in anthracite, but the higher the percentage above this value, the greater the ease of ignition; (2) the size of the dust particles; (3) the density of the dust cloud—since a self-sustained explosion can occur only between certain upper and lower limits of dust-and-air mixture, limits which vary with the different fuels; (4) the intensity and total energy of the source of ignition; and (5) the degree of confinement. If the coal is dried before pulverizing it should not be done by the direct passage of flame or very hot gases over it, as this may result in smouldering masses which may produce explosions in the grinding machinery; an indirect method of drying is safer. The transportation systems, pulverizers, classifying separators, etc., should be dust tight, and bins should be fully enclosed, except for vents. Free dust should not be allowed to accumulate, and should be swept up only after wetting, or mixing with an excessive amount of incombustible material; or, preferably, it should be removed by means of a vacuum system. If all surfaces in the preparation plant are of light color, the deposits of dust are more readily seen. All possible sources of ignition should always be eliminated. Switches should be in dust-proof enclosures, or outside of the preparation building. Sparks from motors, loose sockets, etc., should be guarded against. Rapid moving machinery should be grounded to prevent discharges of static electricity.[26]

The installation and operation of pulverized fuel systems should conform to the safety Regulations of the National Board of Fire Underwriters, of New York City, for the Installation of Pulverized Fuel Systems, as approved by the American Standards Association. This publication also contains recommended procedure for the extinguishing of fires in various parts of pulverized coal systems.

431. Ash and Dust Problems.—(a) From 10 to 40 per cent of the ash of pulverized coal is deposited in the furnace bottom, the remainder being carried by the exit gases as fine dust. The satisfactory disposal of both the deposited and floating ash has constituted one of the most serious problems connected with the use of powdered coal.

(b) In *furnaces with water screens or water-cooled bottoms*, the ash-pit deposit is in granular form and is readily removed, especially if the furnace bottom is hopper-shaped with sides inclined 45 deg. or more from the horizontal. Cooling by water screens is independent of the thickness of the deposit, whereas the cooling effect of a water-cooled bottom becomes reduced if the ash is allowed to accumulate in a thick layer. The granular ash may be disposed of by discharging it into a water sluice, which carries it to low ground to be filled, or to a settling basin from which it may be removed by a clamshell bucket hoist and loaded into trucks or cars; it may be withdrawn into a water trough from which it is removed by a drag conveyor; or it may be discharged, in a wetted condition, directly into a truck or car placed under the bottom gates of the hopper.

(c) In furnaces having *slag-tap bottoms* the ash is deposited in a molten condition, and the liquid is tapped off once or twice a day, usually into a water spray by which it is chilled and disintegrated into small particles that are sluiced away. This method of ash disposal is particularly applicable for use with ash which fuses at low temperature; but to maintain the deposit in liquid form, the burners must be of suitable type and location.

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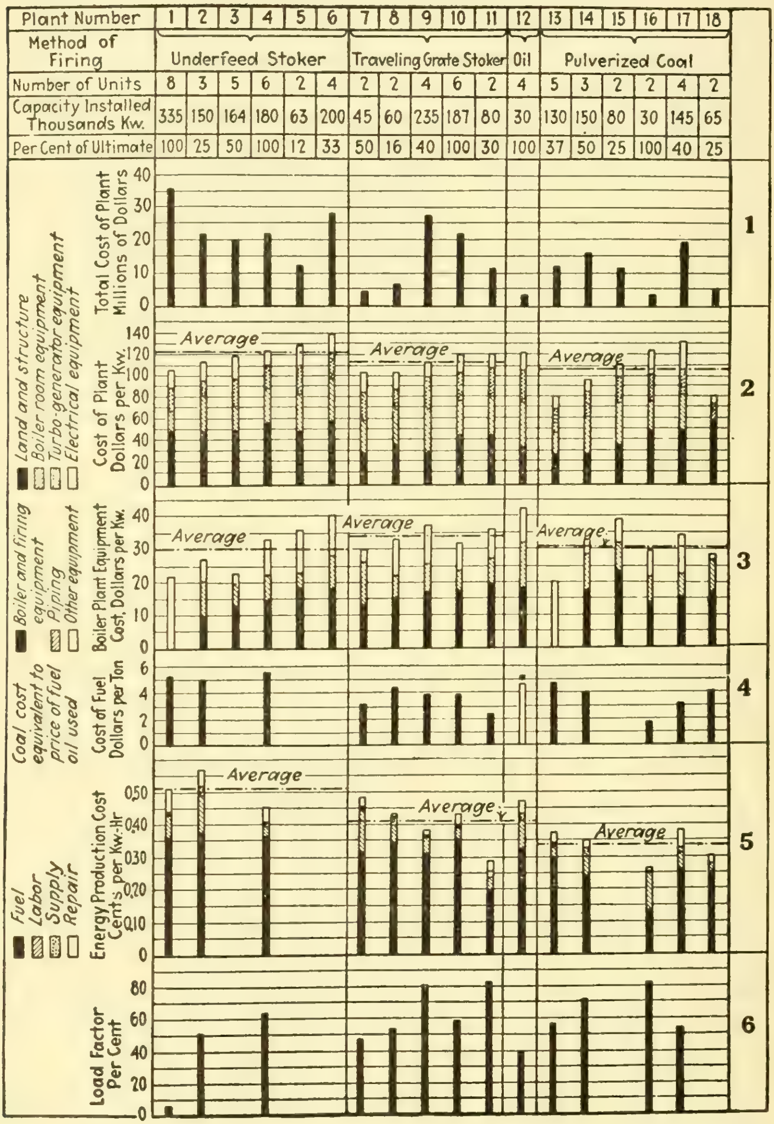
[26] References: "Dangers of Explosion from Coal Dust," by H. P. Greenwald, Lefax, 3289, March, 1928. "Explosion Hazards from the Use of Pulverized Coal," by L. D. Tracy, U. S. Bu. of Mines Bul. 224. *For additional references, see p.* 558.

(d) The **fly-ash,** as already noted, may cause erosion of furnace linings, clog the gas passages, and constitute a serious nuisance when discharged from the stack. The first difficulty is overcome by using suitably designed furnaces; the second by properly arranging the gas passages through the boiler and other parts, and by providing means for readily



Plant Number	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Method of Firing	Underfeed Stoker						Traveling Grate Stoker					Oil	Pulverized Coal					
Number of Units	8	3	5	6	2	4	2	2	4	6	2	4	5	3	2	2	4	2
Capacity Installed Thousands Kw.	335	150	164	180	63	200	45	60	235	187	80	30	130	150	80	30	145	65
Per Cent of Ultimate	100	25	50	100	12	33	50	16	40	100	30	100	37	50	25	100	40	25

FIG. 623.—Illustrative Cost Data for Stoker, Oil, and Powdered-Coal Firing.

removing deposits from heat-transmitting surfaces and from collecting pockets. The removal of ash dust from the gases is accomplished by means of mechanical dust catchers and gas washers (as described in Sect. **391** (d)), which, however, are often not wholly satisfactory unless two or more are used in series. Cottrell electrostatic precipitators have been used successfully to remove from 80 to 95 per cent of the dust, but they occupy large space, cost from 1 to 3 per cent as much as the rest of the plant, and use about 0.03 per cent of the plant's power output. Much effort is being devoted to develop new and better ways of solving the problem of dust removal, not only from the stack gases but also from the air issuing from vents in the coal-preparation system.

432. Costs and Selection.—(**a**) Table LIII gives the detailed costs of powdered-coal preparation in several typical plants; and other cost data are shown in Fig. 623, for plants that are oil- or stoker-fired and for those using pulverized coal.[27] Although the cost of coal preparation and feeding is a little higher with powdered coal than with stoker firing, the item is small, and the total cost of the energy generated by the station may be appreciably lower with the former than with the latter, as is shown in line 5 in the figure. Also, the plant using pulverized coal may cost less, if anything, than the stoker-fired plant, as shown by line 2. The distribution of energy consumption, kw-hr. per ton of fuel burned, is indicated in Fig. 624, for the Calumet unit using the bin system.[28]

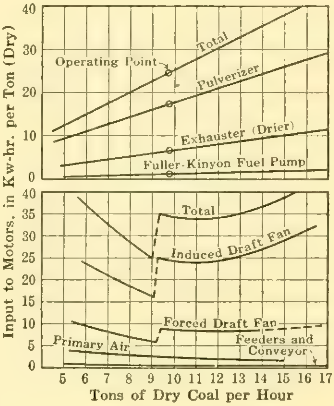


Fig. 624.—Energy for Preparing and Burning Pulverized Coal.

(**b**) The data given are merely illustrative. In an actual project comparisons should be made with the different fuels available; and the various items given in Sects. **420** (**a**) and **421** (**b**), (**c**), and (**d**), may have to be analyzed and compared before the best fuel, burning system, and arrangement of apparatus can be selected for a given set of conditions. In some cases only a few of the items are determining ones, and little attention need be given to the others.

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[27] This Table and Fig. 623 are from "Powdered Coal for Steam Purposes," by Kreisinger and Frisch, before the World Power and Fuel Conference in London in 1929. Table and figure were reproduced in Power, January 15, 1929, p. 100.

[28] Power Aug. 7, 1928, p. 227; also Proc. N.E.L.A., 1928, p. 1424.

TABLE LIII

COAL PREPARATION COSTS IN TYPICAL PULVERIZED COAL PLANTS

(Data Taken from Prime Movers Committee Reports, N.E.L.A.)

Company	Cleveland Electric Illum. Co.	Detroit Edison,	Duquesne Light Co.	Illinois Electric Power Co.	Pennsylvania Water Co.	Colorado Power and Light Co.	Sioux City Gas and Electric Co.	Union El. Lt. and Power Co.	Wisconsin Traction Co.	Texas Power and Light Co.
Plant	Lakeshore	Trenton Channel	Colfax	Peoria	Holtwood	Valmont	Sioux City	Cahokia	Appleton	Trinidad
Coal milled, tons per year	259,821	423,906	307,157	138,675	47,234	117,244	54,540	427,064	63,105	126,541 (6 months)
Tons of coal per mill hour	5.26	6.71	.....	6.75	6.89	6.3	7.41	.....	5.86	.....
Kind of coal	West Va. Bituminous	West Va. Bituminous	Pittsburgh Bituminous	Illinois Bituminous	Bituminous and Anth.	Colorado Lignite	Midwestern Bituminous	Illinois Bituminous	Illinois Bituminous	Texas Lignite
COSTS:										
Energy, cents per ton	9.0363	7.97	.....	8.81	.....	.....	9.83	7.29	14.22	.....
Operating labor, cents per ton	4.5650	7.08	.....	3.25	23.2	5.8	16.4	10.35	9.79	10.96
Operating material, cents per ton	0.3618	0.93	.....	0.88	4.7	0.5	0.52	.....	1.02	0.76
Maintenance labor, cents per ton	2.9830	2.74	1.46	3.48	6.0	1.0	3.8	11.05	2.78	1.45
Maintenance material, cents per ton	5.4653	3.59	1.83	4.76	23.6	0.3	4.76	.....	2.73	.013
Total preparation costs, cents per ton	22.4114	22.31	.....	22.28	.....	.....	35.31	28.69	30.54	.....

CHAPTER XXXIII

THE BURNING OF GAS IN FURNACES

433. Introduction.—(a) The use of gas in furnaces has rapidly increased in importance in recent years. In the public utility power plants alone the annual consumption from 1920 to 1930 has increased about 400 per cent, as compared with a gain of only 15 per cent in the use of coal and a decrease of about 30 per cent in the consumption of oil annually by such plants in the same period. During 1930, gas was used in producing approximately 10 per cent of the electricity generated in steam power plants of public utility companies. Gas is also used extensively as a fuel in heating installations, in metallurgical and heat-treating furnaces, in engines, and for domestic purposes.

The gaseous fuels which are burned in furnaces include natural, coke oven, producer, blast furnace, water, oil, and acetylene gas; but, at present, natural gas is the one most widely used for such purposes.

(b) The sources, typical analyses, and heating values of the most important of the gaseous fuels are given in Sect. **368**. The limits of inflammability of gases and vapors are discussed in Sect. **379**; and Table XLI gives the values of the upper and lower limits of inflammability for many different *single* gases and vapors. The inflammability limits for *mixtures* can be obtained by means of Eq. (532) when used in connection with this table. The speed of flame propagation was discussed in Sect. **380**. Other considerations that affect the design and operation of burners and furnaces using such fuels are the ignition temperature of the gas and factors influencing flame velocity, and these will now be considered.

434. Ignition Temperatures of Gases.—(a) The ignition temperature of a gas is the lowest temperature at which the rate of energy liberation due to the combustion reactions just exceeds the rate of energy loss so that combustion becomes self-propellent. The ignition temperature is not a property of the gas alone, but depends upon a great many other factors, such as the proportion of combustible gas in the inflammable mixture, the other gaseous constituents of the inflammable mixture, the pressure of the mixture, the method of heating, and the size and shape of the containing vessel. In general, the ignition temperature is lower in oxygen than in air and is lowered by increasing the pressure of the mixture.

(**b**) The ignition temperatures of some common gases, as determined by Dixon and Coward by slowly heating the combustible in a stream of air at atmospheric pressure, are given in Table LIV.

Since the experiments yielded a range of temperatures, instead of a single temperature, the ignition temperature is clearly not a property of the fuel alone. Under the conditions of the experiment, thorough mixing of the combustible gas with air was impossible and this affected the temperature of ignition.

TABLE LIV *

IGNITION TEMPERATURES OF GASES IN AIR AT ATMOSPHERIC PRESSURE

Gas	Ignition Temperature, Deg. Fahr.
Methane (CH_4)	1202–1382
Carbon Monoxide (CO)	1191–1216
Hydrogen (H_2)	1076–1094
Ethylene (C_2H_4)	1008–1018
Ethane (C_2H_6)	968–1166
Acetylene (C_2H_2)	763– 824

* Journal Chem. Soc., Vol. 95, 1909.

435. Factors Affecting Flame Speed.—(**a**) The rate of flame propagation or speed at which a flame travels through a gas-air mixture may be fairly uniform and slow, or there may be an accelerating and rapid explosive wave set up under some conditions of ignition. The speed of uniform movement of the flame in air, or the *normal velocity* of flame propagation, depends largely upon the proportion of gas in the gas-air mixture. The speeds of uniform movement in air of the flames of some individual gases[1] are shown in Fig. 625. The curves are drawn for flame propagation in a tube 2.5 cm. (1-in.) in diameter. The diameter affects the speed of the flame; in general, decreasing the diameter of the tube decreases the speed of the flame for any mixture ratio, largely because of the greater ratio of surface to volume and consequent greater cooling effect of small tubes. The speed of the flame is low for mixtures near the lower limit of inflammability, and increases, at first, as the amount of the combustible gas increases, reaches a maximum, then drops off, and, near the upper limit of inflammability, becomes reduced to a speed nearly as low as that at the lower limit. The mixture which gives maximum flame speed is usually slightly richer in combustible than the gas-air mixture chemically correct for complete combustion. For example, the chemically correct mixture for the combustion of hydrogen contains 2 ÷ 6.78 or about 29.5 per cent hydrogen by volume; whereas, the mixture that gives maximum flame speed contains about 37 per cent hydrogen.

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[1] Bulletin No. 301, U. S. Bureau of Mines, p. 67; "Facts Relating to the Production and Substitution of Manufactured Gas for Natural Gas," by W. W. Odell.

(**b**) The speed of uniform movement of the flame with a *mixture of several gases* may be approximated by calculation, provided there are not large amounts of nitrogen, carbon dioxide, or oxygen in the original gas mixture. For that purpose, it can be assumed that each individual gas-air mixture in the complex gaseous mixture burns independently and with a gas-to-air ratio which gives in it the same speed of flame propagation that the complex mixture has under the same conditions.[2] If, as is commonly the case, the *maximum speed* of uniform movement is desired, the per cent of the complex gas, by volume, in the **gas-air mixture giving maximum speed** may be calculated from the following equation, which is similar in form to Eq. (532):

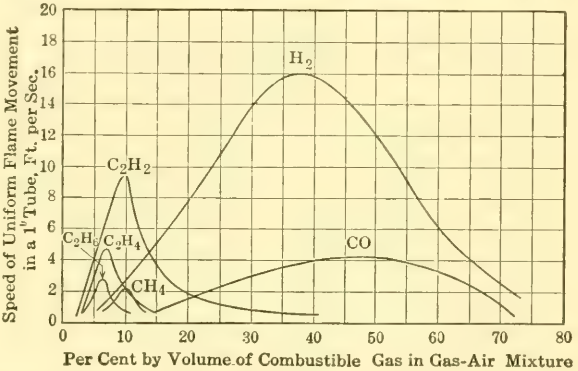

FIG. 625.—Normal Flame Velocities through Various Gas-Air Mixtures in a 1-Inch Tube.

$$M_m = \frac{f_1 + f_2 + f_3 + \ldots}{f_1/M_1 + f_2/M_2 + f_3/M_3 + \ldots} \quad . \quad . \quad . \quad (573)$$

in which

f_1, f_2, f_3 = the fractions, by volume, of the respective constituent simple gases in the complex gas mixture;

M_1, M_2, M_3 = per cents, by volume, of the respective simple gases in their simple gas-air mixtures giving maximum flame speed, as obtained from Fig. 625.

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[2] For more detailed discussion and for the derivation of Eqs. (573) and (574) which follow, see pp. 265 to 275 in "Fuels and Their Combustion" by Haslam and Russell.

For example, for natural gas C of Table XXXV (since the proportions of methane and ethane in the simple gas-air mixtures giving maximum flame speed are 9.8 and 6.4 per cent, respectively) the gas-air mixture giving maximum flame speed is one containing

$$M_m = \frac{0.747 + 0.237}{0.747/9.8 + 0.237/6.4} = 8.7 \text{ per cent natural gas.}$$

(**c**) The **maximum speed** of uniform flame movement for a complex-gas-air mixture is

$$v_m = \frac{av_a + bv_b + cv_c + .}{a + b + c + \ldots} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (574)$$

in which a, b, and c = the fractions, by volume, of the respective maximum-speed simple gas-air mixtures in the complex gas-air mixture giving maximum flame speed.

v_a, v_b, and v_c = the respective maximum flame speeds of the simple gas-air mixtures.

For the natural gas previously considered, the maximum speed of uniform flame movement is

$$v_m = \frac{[(0.747 \times 0.087) \div 0.098]\, 2.2 + [(0.237 \times 0.087) \div 0.064]\, 2.8}{[(0.747 \times 0.087) \div 0.098] + [(0.237 \times 0.087)] \div 0.064]}$$

$$= 2.4 \text{ ft. per sec.}$$

This is the flame velocity in a stagnant mixture.

(**d**) If the burning mixture itself is moving, the *absolute velocity* of the flame depends on the velocities of the mixture and flame relative to each other. Thus if the gaseous combustible mixture just considered were forced through a tube of uniform cross-section at velocity of 2.4 ft. per sec. and were ignited, then the flame would be stationary in the tube and would hover at the place of ignition. If the velocity of the mixture were cut down to 1 ft. per sec. the flame would travel back in the tube against the direction of flow at a rate of about 1.4 ft. per sec.; in other words, the flame would flash back. If the velocity were increased to 4 ft. per sec., the flame would travel in the direction of gas flow at a rate of about 1.6 ft. per sec. and would blow out upon reaching the end of the gas stream.

If, now, an orifice or restriction to gas flow were placed in the pipe, and if the original velocity of gas flow of 2.4 ft. per sec. were maintained in the pipe proper, there would be a cross-section down stream from the orifice where the velocity of 2.4 ft. per sec. would again be found, and if the gas were ignited there the flame would hold at this place. The rate of gas flow could now be cut down, and the flame would move back toward the orifice, where the velocity is higher than the average, but the flame would not flash through this restriction until a considerable reduction is made in flow, the decrease depending on the ratio of orifice area to pipe area.

(e) If the gas-air mixture issues under pressure from the end of a simple nozzle or *burner*, the flow becomes retarded as the stream proceeds. Thus, if the discharge velocity is a little above the rate of flame propagation in a stagnant mixture, the ignition point will be beyond the burner outlet. Lowering the discharge velocity causes the flame to approach closer to the burner, and may result in a flareback if air is mixed with the gas within the burner and pipe leading thereto. Increasing the exit velocity causes the flame to recede from the burner and may result in the flame blowing out.

(f) A *dilution with air* would reduce the flame speed, and wherever the mixture becomes leaner than the lower limit of inflammability (which, for gas C is found from Eq. (532) and Table XLI to be 4.6 per cent by volume) the flame would go out. Hence, when a large part of the combustion air is supplied through the furnace walls or bottom, tuyere blocks or tunnels should be placed just beyond the burner nozzle or orifice to prevent excessive localized dilution and to confine the mixture to its proper proportions. These blocks, when hot, also aid the combustion by heating the mixture and producing turbulence.

In addition to the properties of the gas mixture already mentioned, the flame, and therefore the burner design and operation, are also influenced by other complex factors depending on the heat transfer, temperature, turbulence and furnace draft.

436. Gas Burners.—(a) As the types and makes of gas burners are almost innumerable, only the general principles of such burners will be considered and but a few typical illustrations given.

Gas burners for use in furnaces may be classified according to the gas and air pressures employed, according to whether refractories are used in them or not, and in various other ways; but one of the most important distinctions is the degree of mixing of the gas and air that takes place within the burner. In general, burners that give but little mixing produce long luminous flames, while those that discharge an intimate mixture of gas with all the combustion air required, produce short non-luminous flames.

If, as in metallurgical furnaces, high flame temperatures are required, the air supplied for combustion must be preheated to a very high temperature. If the hot air and gas were mixed within the burner, there would be danger of ignition therein; hence, in such cases, the burner supplies only the gas, and the mixing takes place within the furnace. In such applications a simple "pipe burner" is commonly used.

Most gas burners used in boiler furnaces give some premixing of air and gas within the burner. These burners are usually of the aspirating type, i.e., the flow of one fluid is used to induce the desired flow in the other. One design of burner that gives a moderate amount of mixing of the gas and air is shown in Fig. 626. Burners of this kind differ as to the way the gas is introduced into the stream of air; in some it is introduced in various ways around the outside of the stream, and in others it is injected into the center thereof. Fig. 604 shows a burner in which the gas issues in a thin film around the periphery of the burner throat, through which passes a rotating stream of air supplied from the registers. This burner is also arranged to use oil and powdered coal; thus, as the conditions may demand, any combination of the three fuels can be burned simultaneously or any one fuel can be used separately. The burner may be operated either with natural draft or with forced draft.

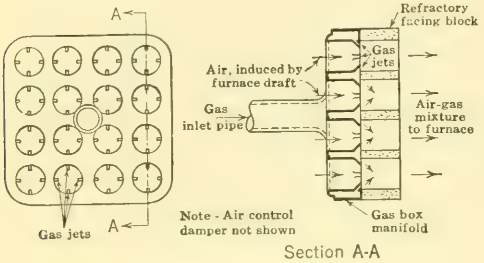
A
A
Gas jets
Refractory facing block
Gas jets
Air, induced by furnace draft
Gas inlet pipe
Air-gas mixture to furnace
Note - Air control damper not shown
Gas box manifold
Section A-A

FIG. 626.—Gas Burner with Moderate Mixing.

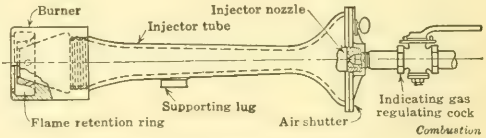

FIG. 627.—Venturi Type of Nozzle.

Rapid-mixing burners are often of the Venturi type with central gas-injector nozzles, one example of which is shown in Fig. 627. The reduced pressure at the restricted throat induces the flow of the primary air through the burner, and causes rapid mixing. The flame-retention ring maintains a small flame to reignite the main jet when blown out. The gas is supplied to burners of this type under pressures up to as high as 20 lb. per sq. in. In the modified venturi burner shown in Fig. 628 the mixing is done in two stages.[3]

Separate gas burners are often installed in furnaces primarily designed to use other kinds of fuel, so as to be available if it is necessary to use gas to supplement or replace the fuel normally burned. Gas burners are used also for the initial ignition of pulverized coal. In this case they are slotted so as to produce a fanlike flame at the mouth of the coal burner, and this flame is directed at right angles to the path of coal flow so that ignition is practically instantaneous.[4]

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[3] For further discussion see "Natural Gas Burner Applications," by E. B. Plapp, in Combustion, March, 1930, p. 24; "Better Boiler Room Operation," by Otto de Lorenzi, in Power House, March, 1930; annual reports of Committee on Burning of Liquid and Gaseous Fuels, N.E.L.A. Proc.

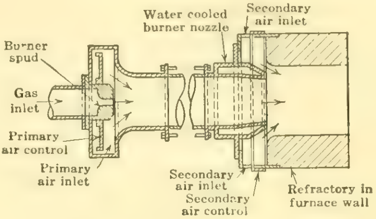

Fig. 628.—Rapid-Mixing Burner with Separate Inlets for Primary and Secondary Air.

437. Gas-Fired Furnaces.—Gas-fired furnaces involve very few special considerations in design. As in the pulverized-coal and oil-fired furnaces, a sufficient distance for flame travel must be allowed to insure complete combustion before the furnace gases meet any heat-absorb-

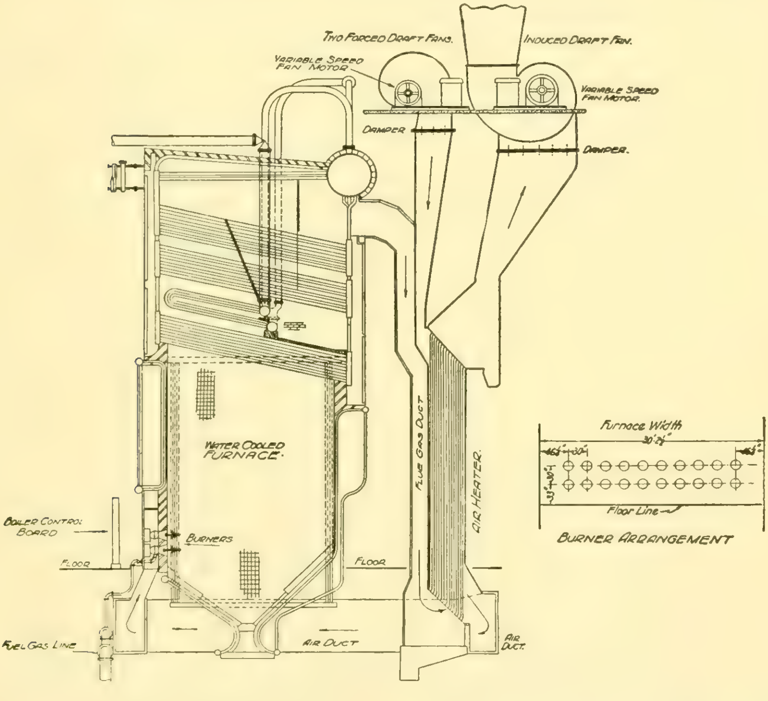

Fig. 629.—Unit Fired with Oil and Gas. Surfaces, Sq. Ft.: Boiler, 34,162; Superheater, 6,533; Water Walls, 3,240; Air Heater, 49,000. Furnace Vol., 19,800 Cu. Ft.

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[4] See Power, June 9, 1931, p. 912.

ing surfaces. The energy liberation rates allowable in gas-fired furnaces are about the same as in the oil-fired furnaces; the average rate in use, at the present time, is from 20,000 to 40,000 B.t.u. per hr. per cu. ft. of furnace volume. Where rapid-mixing gas burners are used and there is no attempt to produce soot in the flames by adding benzene or by other methods,[5] the flame in a gas-fired furnace is nearly nonluminous and has very low emissivity. As a consequence of this and of the moderate flame temperatures, there is a smaller proportionate amount of heat transferred by radiation in the gas-fired than in the oil-fired furnace. Although this reduces refractory troubles, still, if low exit gas temperatures are to be secured, there must be proportionately more heat transferred to the water and steam by convection. Since the rate of heat transfer by convection is greatly influenced by the velocity of gas flow, the passages in the boiler should be so proportioned as to produce fairly high furnace-gas velocities. To provide for the possible substitution of oil as the fuel, the furnaces are commonly designed to be suitable for either of these fuels, as in Fig. 629.

438. Comparative Performances.—Pulverized coal, oil, and gas are frequently burned in the same boiler furnace, and test results are available showing the efficiencies of the same steam-generating unit when burning the different fuels. In general, in the same boiler furnace, as it is at present designed, and at the same load, gas is usually burned with a slightly lower efficiency than oil. As an illustration, the steam-generating unit shown in Fig. 629 was tested[6] with constant feed-water temperature and the same excess air when burning fuel oil and also when burning the natural gas C of Table XXXV. The efficiency of the unit when using each fuel is shown in Fig. 630 as a function of the load, expressed in terms of the unit rate of heat absorption in thousands of B.t.u. per hr. per sq. ft. of heating surface. A comparison with the results obtained with coal might be even more unfavorable for the gas, unless poor coal is used.

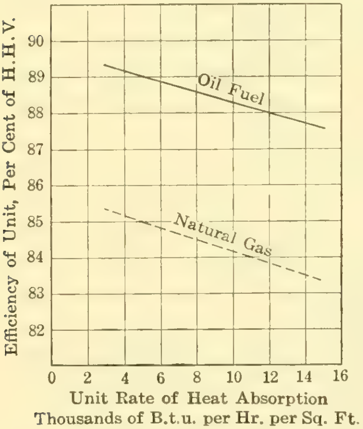

FIG. 630.—Comparative Efficiencies with Oil and Gas Firing.

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[5] See p. 280.

[6] Proc. N.E.L.A., 1930, p. 866. Comparison of test results, Boiler 50, Long Beach Steam Plant No. 3, Southern California Edison Co.

The reason for this difference of efficiency in favor of the oil or coal firing is best explained by investigating the combustion characteristics of the carbon-hydrogen components of fuels.

439. The Effect of the Hydrogen Content of a Fuel upon the Efficiency of a Steam-Generating Unit.—(**a**) The hydrogen percentage in fuel burned in a steam-generating unit affects appreciably the losses and efficiency of the unit, and in amounts that can be readily approximated by computations based on the assumption that the fuel is a mechanical mixture of carbon and hydrogen.

(**b**) On this basis, the **higher heating value** of such a fuel, in terms of the weight of hydrogen (w_H) in one pound of the fuel, is

$$\text{H.H.V.} = 14{,}150(1 - w_H) + 61{,}000w_H = 14{,}150 + 46{,}850w_H \quad . \quad (575)$$

This equation is represented graphically in Fig. 631. Actually, commercial fuels are not mechanical mixtures of carbon and hydrogen; therefore, their heating value is not given exactly by this equation, because the heats of combination of the hydrocarbons are not considered. However, the error is negligible for the purpose of these calculations.

(**c**) The **weight of the dry products of combustion** and of the water vapor in these products may be calculated readily from the chemical reaction equations. In terms of the weight of hydrogen in one pound of the hydrocarbon fuel, the *weight of the dry products* of combustion per pound of fuel completely burned with an excess air fraction equal to x, is

$$\begin{aligned} w_g &= 12.5(1 - w_H) + x \times 11.5(1 - w_H) + 26.5w_H + x \times 34.5w_H, \\ &= 12.5 + 14w_H + 11.5x + 23xw_H \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (576a) \end{aligned}$$

For example, if $x = 0.15$,

$$w_g = 14.225 + 17.45w_H \quad . \ . \ . \ . \ . \quad (576b)$$

The **weight of water vapor formed** from the complete combustion of the hydrogen in one pound of the fuel is

$$w_{vH} = 9w_H \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (577)$$

Equations (576*b*) and (577) are represented graphically in Fig. 632.

(**d**) If the exit gas temperature be taken as t_x, and the temperature of the entering dry air and fuel as t_a, then the **dry gas loss** is, in B.t.u. per pound of fuel,

$$Q_d = 0.24w_g(t_x - t_a) \quad . \ . \ . \ . \ . \ . \ . \quad (578a)$$

With $t_x = 400$ deg. fahr., $t_a = 80$ deg. fahr. and $x = 0.15$, then from Eqs. (576*b*) and (578*a*),

$$Q_d = 0.24(14.255 + 17.45w_H)(400 - 80) = 1090 + 1340w_H \, . \quad (578b)$$

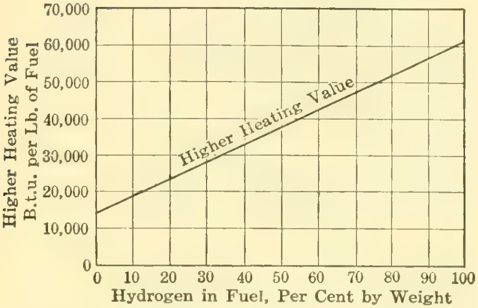

FIG. 631.—Higher Heating Value vs. Hydrogen Percentage in Fuel, by Weight.

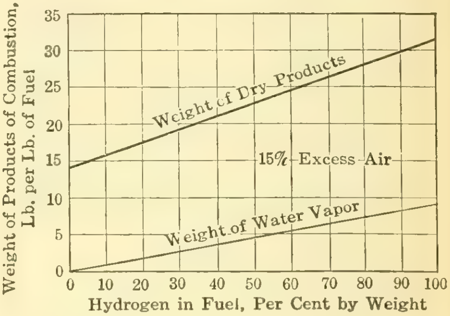

FIG. 632.—Weights of Products of Combustion vs. Hydrogen Percentage in Fuel, by Weight.

The **fractional dry gas loss,** expressed as a fraction of the higher heating value of the fuel, becomes

$$Q_d/\text{H.H.V.} = \text{Eq. } (578a) \div \text{Eq. } (575) \quad . \; . \; . \; . \; . \; . \; . \quad (579a)$$

and, for the conditions assumed

$$Q_d/\text{H.H.V.} = (1090 + 1340w_H)/(46{,}850w_H + 14{,}150) \quad . \; . \quad (579b)$$

Similarly, the **loss of heat due to the water vapor** formed from hydrogen and present in the products of combustion, in B.t.u. per lb. of fuel, originally at temperature t_f, is from Eqs. (577) and (567*b*),

$$Q_h = 9w_H(1089 + 0.46t_x - t_f) \quad . \; . \; . \; . \; . \quad (580a)$$

and, for the assumed conditions, and with $t_f = 80$ deg. fahr.

$$Q_h = 9w_H(1089 + 0.46 \times 400 - 80) = 10{,}737w_H \quad . \quad (580b)$$

The **fractional loss due to moisture** is

$$Q_h/\text{H.H.V.} = \text{Eq. } (580a) \div \text{Eq. } (575) \quad . \; . \; . \; . \; . \quad (581a)$$

and, for the assumed conditions,

$$Q_h/\text{H.H.V.} = (10{,}737w_H)/(46{,}850w_H + 14{,}150) \quad . \quad (581b)$$

The loss percentages represented by Eqs. (579*b*) and (581*b*) are plotted in Fig. 633 against the hydrogen content of the fuel. The sum of these losses is also shown, and it increases as the hydrogen in the fuel increases. If combustion is complete with 15 per cent excess air, if there is no loss of heat in the refuse from the fuel and no loss of heat by heat transfer, if the exit gases leave the last heat-absorbing surface of the steam-generating unit at 400 deg. fahr., if dry air and dry fuel are supplied at 80 deg. fahr., and if the fuel is a mechanical mixture of carbon and hydrogen, then by subtracting the total loss of energy in the products of combustion from 100 per cent, the efficiency of a steam-generating unit, operating under the imposed conditions, is secured. The upper curve in Fig. 633 shows the manner in which the efficiency of the steam-generating unit decreases under these conditions as the hydrogen in the fuel increases. Note, for example, that the efficiency of the unit is 92.3 per cent if the fuel is pure carbon, and only 78.4 per cent if the fuel is pure hydrogen.

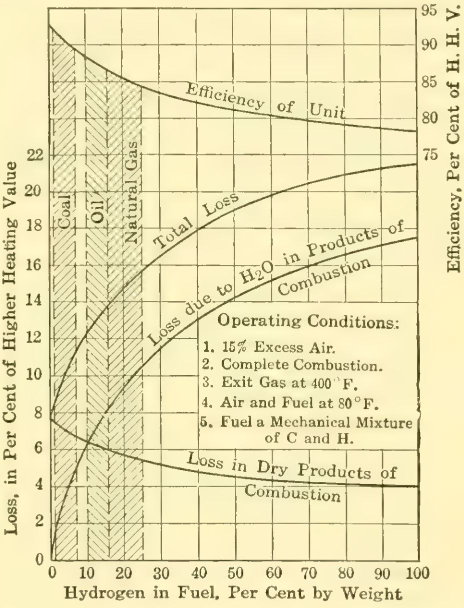

Fig. 633.—Effect of Hydrogen Percentage in the Fuel on the Losses and Efficiency of a Steam-Generating Unit.

Obviously, the operating conditions used in this discussion are not met in all steam-generating units; nevertheless the discussion shows the general effect of the hydrogen content of the fuel upon the possible efficiency of a unit. To prove that fair assumptions have been made, compare this calculated performance with the test performance discussed in the preceding section. At a heat-absorption rate of 5000 B.t.u. per hr. per sq. ft., the actual steam-generating unit when fired with oil gave an efficiency of 89 per cent, and when fired with natural gas an efficiency of 85 per cent was secured at the same load and operating conditions. The oil fuel used contained 10.94 per cent hydrogen, by weight, and the natural gas used was 22.6 per cent hydrogen, by weight. The calculated efficiencies for the prescribed conditions are

87.5 per cent for the oil and 84.7 per cent for the natural gas, showing a close correlation between the calculated and test performances.

(e) For making a comparison of results with coal, oil and gas (natural) the approximate ranges of hydrogen content of these three fuels have been indicated in Fig. 633. For the given operating conditions, these ranges show that a steam-generating unit should give an efficiency of around 90 per cent when burning coal, 87 per cent with oil, and 85 per cent with natural gas. It should be emphasized, however, that the calculated values will be modified by variations in the temperatures of exit-gas, air or fuel, and in the excess air and incompleteness of combustion, by loss of heat in the refuse from the fuel, by heat transfer, and by the heats of combination of the hydrocarbons; but the *relative* performances with different fuels should be similar to these results *when the operating conditions are the same with all the fuels.* It should be noted, however, that the excess air requirements of the different fuels are not quite the same. In general, gas and oil can be burned with less excess air than coal without excessive loss due to incomplete combustion. This produces an effect upon efficiency in favor of gas over oil, and of oil over coal,[7] and is not considered in this analysis, —for the excess air is assumed to be 15 per cent in all cases.

440. Flame Luminosity and Its Effect.—Natural gas burns with only an extremely slight luminosity, hence transmits by radiation to the walls and boiler tubes exposed in the furnace a smaller proportion of the heat evolved than does an oil flame, which is highly luminous due to the presence of small particles of carbon formed during the "cracking" of the hydrocarbons and from which the radiation approaches that of the black body, as explained in Sect. **337.** Thus, with gas as the fuel, a greater percentage of the heat liberated remains in the flue gas and is transmitted by convection to the heat-absorbing surfaces of the boiler, superheater, and recovery apparatus, than when oil is used in the same unit under the same conditions. The effect of the difference in the distribution of heat transfer when burning gas and oil is clearly illustrated by the steam temperatures obtained with convection superheaters in the tests of the Long Beach unit already referred to in Sect. **438**. The

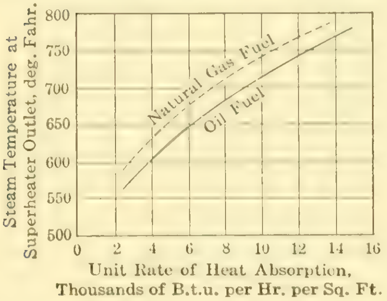

FIG. 634. Comparative Steam Temperatures from Convection Superheaters with Natural Gas and Oil Fuel.

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[7] See Mech. Eng., July, 1926, p. 703.

steam temperatures obtained with the two fuels burned in the water-cooled furnace of this unit, with 15 per cent excess air and with feedwater at 360 deg. fahr., are shown in Fig. 634.

Thus, the luminosity of the flame in the furnace affects the performance of the various parts of the unit, and its influence should be considered when designing a new installation.

441. Surface Combustion and Submerged Combustion of Gas—(a) The so-called **surface combustion** of gas consists in burning gas, without flame, in contact with an incandescent solid surface. All surfaces possess, in varying degree, the power of accelerating gaseous combustion. At low temperatures, metallic surfaces of platinum, ferric oxide, and nickel oxide are the best catalysts; at high temperature, incandescent coal and refractory materials are very satisfactory. In commercial applications of surface combustion, a homogeneous mixture of gas and air in the proper quantitative proportions for complete combustion is burned in contact with a localized surface of a solid so that a condition of incandescence is maintained at this surface. The advantages of this system are accelerated rates of complete combustion with minimum excess air, and rapid rates of heat transfer.

Application of surface combustion in a steam boiler has been made by Professor W. A. Bone and Mr. C. D. McCourt in the Bonecourt boiler. The fire tubes of the boiler are filled with porous refractory material through which the gas-and-air mixture is drawn by a fan at such rate that the combustion is localized in the desired region in the filling. In these tubes, energy has been liberated at a rate of 700,000 B.t.u. per hr. per cu. ft. of combustion space, and efficiencies as high as 94 per cent have been secured when an economizer was used.

As applied to metallurgical furnaces, the burners may direct the gas-air mixture onto a bed of incandescent refractory granules from which the heat is radiated usually to a roof and thence to the material being heated. In another method, the gas-air mixture is directed by the burner into a cement-lined tunnel so proportioned that the combustion of the gas is nearly completed while it is in contact with the incandescent tunnel surface; the hot gases then sweep the refractory lining of the furnace, transferring heat to this lining largely by convection.[8]

(b) **Submerged combustion** of a gas can be secured by installing a gas burner, with or without a combustion chamber, directly in a liquid bath. High gas velocities are used in order to prevent back firing of the flame. In the latest designs, the gas and air are thoroughly mixed and directed into a refractory-lined combustion chamber; ignition is by an electric spark. The products of combustion are discharged from the combustion chamber into the liquid bath; extremely minute bubbles of these gaseous products ascend up through the liquid and a large surface of contact is exposed to give very effective heat transfer.

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[8] For additional information on surface combustion, see "Coal and Its Scientific Uses," by W. A. Bone, Longmans, Green & Co.; "Fuels and Their Combustion," by Haslam and Russell, p. 307; "Surface Combustion," by A. E. Blake, Proc. Engrs. Soc. Western Pa., Vol. 36, 1920, pp. 175–204.

Submerged combustion has been successfully applied to the evaporation of liquids such as phosphoric and sulphuric acid, to the melting of white metals, to cleaning and dyeing, and to water heating. Since the products of combustion are mixed with any generated steam in almost equal proportions, and since the compressor units must compress the gas and air to a pressure above that maintained in the pressure vessel, submerged combustion has not been widely applied in power plants, although the delivered steam and gas mixture might be used in an engine or turbine.[9]

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[9] For additional information on submerged combustion, see "Submerged Combustion," by Norman Swindin, World Power, Sept., 1927; "Submerged Combustion," by David Brownlie, Combustion, June, 1930, p. 37; Power Plant Engrg., Sept. 1, 1929, p. 944.

ADDITIONAL REFERENCES

"Radiation from Luminous and Non-Luminous Natural Gas Flames," by R. A. Sherman, Trans. A.S.M.E., 1934, IS-56-1, p. 177.

"Gas Engineers' Handbook," by F. Wills and others. McGraw-Hill Book Co., N. Y.

ADDITIONAL REFERENCES ON THE BURNING OF POWDERED COAL

"Burning Characteristics of Pulverized Coals and the Radiation from Their Flames," by R. A. Sherman, Trans. A.S.M.E., 1934, FSP-56-6, p. 401.

"Correlation of Grindability with Actual Pulverizer Performance," by M. Frisch and G. C. Holder, Combustion, June 1933, p. 29.

"Pulverized Fuel-Burning Experience at Buzzard Point Station," by H. G. Thielscher, Trans. A.S.M.E., 1934, FSP-56-18, p. 885.

"Mill Drying at Lakeside," by M. E. Fitze, Combustion, June 1934, p. 22.

"Pulverized Anthracite as Boiler Fuel," Combustion, Sept. 1934, p. 10.

"Burning Pulverized Lignite, by R. F. Throne, Mech. Eng., June 1934, p. 325; discussion Sept. 1934, p. 564.

CHAPTER XXXIV

THE BURNING OF OIL IN FURNACES

442. Introduction.—(**a**) Oil is burned extensively for the purpose of generating steam in steamships, locomotives, industrial and central station power plants, and in heating systems. Approximately half of the Nation's production of crude oil is consumed for such purposes, and of this portion steamships use about 40 per cent, locomotives about 30 per cent; while industrial power plants, central stations and heating systems account for from 4 to 10 per cent each. Oil is also widely used as fuel in many metallurgical furnaces.

(**b**) The **principal advantages of using oil** as fuel in a furnace, are as follows:

(1) Low combustion losses, due to complete combustion with low excess air.
(2) High capacity.
(3) Great flexibility to meet changes in loads.
(4) Low attendance and labor costs.
(5) Less storage space than required with coal, due to greater heating value on either the volume or weight basis.
(6) No deterioration of fuel during storage.
(7) Small amount of refuse.
(8) Small banking losses.
(9) Small draft required.

(**c**) The chief **disadvantages** of oil fuel as compared with gas or coal may be the higher cost of the oil per million B.t.u. of heating value, and the greater uncertainty of the supply at the place where the fuel is to be consumed. These disadvantages limit at present the extensive economical use of fuel oil in land plants to parts of this country near the California, Mexican, Gulf, and Mid-Continent oil fields.

(**d**) The major problems encountered in utilizing oil as fuel, irrespective of the particular installation, are: (1) the storage of the oil, (2) the maintenance of a uniform flow of oil of suitable viscosity to the furnace, (3) the atomization of the oil, and (4) the mixing of the oil with the air requisite for combustion.

The properties of petroleum and petroleum products important in combustion have been discussed in Sect. **359** to **365** incl. The following sections will be devoted mainly to the methods and apparatus used in burning oil in furnaces.

443. Equipment for Handling Oil Fuel.—(**a**) The diagrammatic layout of a typical system necessary in a power station to store and deliver the oil is shown in Fig. 635. The atomization of the oil and the mixing with the combustion air are effected by the burners, which will be discussed in Sect. **445**. The oil-storage and delivery equipment usually consists of (1) a storage tank, (2) strainers, (3) a pump, (4) an air cushion tank, and (5) heaters.

(**b**) The function of the **storage tank** is to provide a reserve supply of oil sufficient to meet the requirements over the longest probable period of time between deliveries. For stationary plants, the tanks are usually made of steel or concrete and they are frequently located underground to guard against fire. The top of the tank should be lower than any point in the oil distribution system, in order to prevent the out flow of oil by gravity from a break in this system. The oil suction pipe extends to within six or eight inches of the bottom of the tank and is surrounded by steam coils by means of which the temperature of the oil may be increased (viscosity decreased) sufficiently for pumping. It is unnecessary to heat all the oil in the tank, so the coils are placed as shown. The vent pipe should be of sufficient size to allow air and vapor to escape rapidly from the tank when it is being filled; this is necessary in order to prevent fluctuations in oil pressure at the burners. The figure also shows the oil fill and the return pipe. The latter returns oils not used at the burners. In addition to the connections shown in

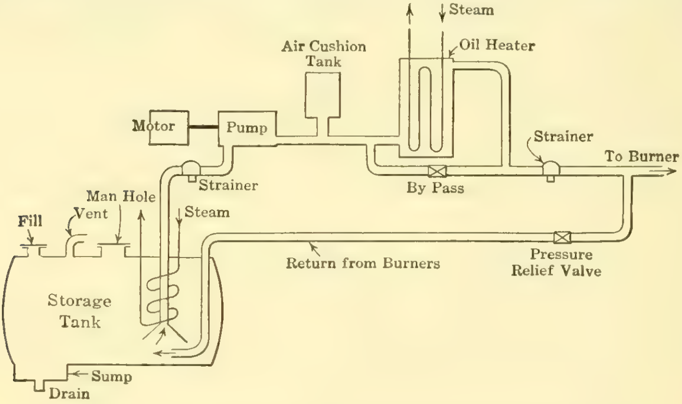

FIG. 635.—Apparatus for Storing and Delivering Fuel Oil.

Fig. 635, the tanks are usually provided with a hand or gear pump to remove water, a smothering pipe through which steam or some smothering gas may be fed in case of fire, and a sounding connection or tank gage to determine the level of the oil.

(c) **Straineers** may be located in the *pump-suction* line or in the oil-heater *discharge* line, or in both, as shown. The suction strainer insures against damage to feed pump and valves, while the discharge strainer removes any sediment such as shreds of pump packing picked up between the pump suction and burner. Frequently an auxiliary strainer is made a part of the burner as a final precaution. The function of this strainer is to remove any remaining sediment which might cause clogging and erosion. The straining elements are usually made in the form of a basket which can be easily removed and cleaned.

(d) The **oil pumps** are located near the storage tank, and raise the pressure of the oil to a value which depends primarily upon the type of burner installed and on the pressure losses that occur before reaching the burner. In medium or large plants, steam-driven simplex or duplex piston or plunger pumps, and electric-driven triplex, screw or rotary pumps have all been used. For smaller installations electric-motor-driven rotary and gear pumps are usually employed. A *pump governor* is needed to maintain the discharge pressure substantially constant; and, since a steady pressure at the burner is very desirable, either air cushion tanks or an elevated oil reservoir, such as a standpipe, are employed to smooth out pressure pulsations.

(e) The function of the **oil heater** is to preheat the oil to a temperature high enough to insure effective atomization at the burner. Either exhaust or live steam may be used as the heating medium, depending upon the necessary oil temperature, which, in turn, depends upon the viscosity of the oil and the type of burner. The types of heaters, or *heat exchangers*, commonly used are as follows: (*a*) Straight-pipe return-tubular, (*b*) straight-pipe multipass, (*c*) double-concentric-pipe coil, and (*d*) film type. Since the controlling resistance to heat transfer between condensing steam and oil is at the oil surface, the overall coefficient of heat transfer realized in such heaters depends primarily upon the velocity of oil flow. Very high rates of heat transfer have been secured in the double-concentric-pipe heaters, and in film heaters of

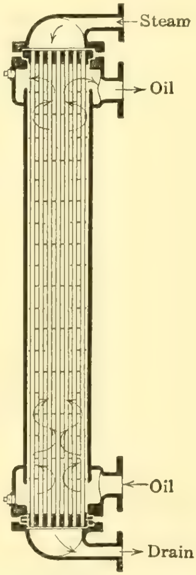

FIG. 636.—Film Type of Oil Heater.

the type in which the oil, flowing at high velocities, is broken up into thin films. One form of film type of oil heater is shown in Fig. 636.

(f) A **by-pass** around the heater is generally used, as shown in Fig. 635, so that **cool oil** may be bled in to regulate the temperature of the oil going to the burner. The **return line** shown carries back to the storage tank the excess oil from each burner. To maintain the necessary pressure at the burner and yet permit circulation when desired, a **pressure-relief valve** is installed in this return line, as shown in the figure. In order to allow for continued operation while cleaning, overhauling or repairing, the storage tanks, pumps and strainers are usually installed in duplicate.

444. The Theories of Oil Ignition.—(a) There are two distinctly different theories concerning the ignition of oil fuel: (1) the "*vaporization*" or "evaporation" theory, and (2) the "*activation*" theory. A complete discussion of these theories is very involved and will not be undertaken in this text; the different viewpoints will, however, be briefly outlined.

(b) The older theory is that **vaporization** or *evaporation* of the liquid fuel must occur before there can be any ignition. It is very probable that some vaporization precedes ignition, because part of the liquid oil evaporates at temperatures below the ignition temperature. The point of uncertainty, however, is whether or not this vaporization is *essential* to ignition. There are many sub-theories under the general vaporization theory: For example, some scientists and engineers believe that the heavy hydrocarbons ($C_{10}H_{22}$, etc.) of the fuel oil are first vaporized and then split up into, principally, ethylene (C_2H_4), methane (CH_4), and acetylene (C_2H_2), and that these gases then ignite and undergo complicated combustion reactions. In support of this theory, Mr. William F. Joachim[1] suggests that at the surface of a small oil globule, the relative proportion of fuel to oxygen is too rich for combustion, and that the proper chemical proportions are not realized until some oil molecules have been released from the oil surface by vaporization and have traveled well into the surrounding gas layer.

(c) The contrary and later theories have the hypothesis that ignition of the liquid phase may and does take place without the *necessity* of previous vaporization. One such theory stresses **activation** or the formation of activated moloxides (unstable hydrocarbon peroxides) on the surface of the liquid oil drops. These moloxides are assumed to decompose with the liberation of sufficient energy to cause the ignition of the remaining hydrocarbons. In short, the moloxides act as primers to the ignition and combustion of the liquid drops. Evidence in support of this theory, which has been accumulated by studies of the ignition and combustion of oil in the cylinders of oil engines, will be briefly discussed in Sect **449** (**c**).

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[1] "Combustion in High Speed Oil Engines," Trans. A.S.M.E., 1929, OGP–51–11, p. 113.

445. Oil Burners for Use in Furnaces.—(**a**) The function of an oil burner is not to burn the oil but to *prepare* it for rapid and complete combustion and to project it into the furnace. Although the exact nature of the ignition and combustion of the oil is uncertain, as explained in the preceding section, the best way to insure complete burning with low excess air is to break up the oil into very minute particles. Oil burners are therefore designed to *atomize,* and in some instances to *vaporize* the fuel.

The **main requirements of an oil burner** are that it (1) atomize or vaporize the oil completely, and be free from clogging or "drooling"; (2) produce a jet of such shape that all parts can be readily mixed with adequate air for combustion; (3) give complete combustion with minimum excess air over the entire range of loading; and (4) provide accessibility and require a minimum of attention and cost for maintenance.

There are a great many different designs and makes of oil burners. Only the general principles and a few illustrations of typical forms will be considered.

Vaporizing burners are used extensively only in relatively small furnaces, such as forging or heat-treating furnaces, where high stack temperatures are available to preheat the air used in vaporizing the fuel. They will not be considered further.

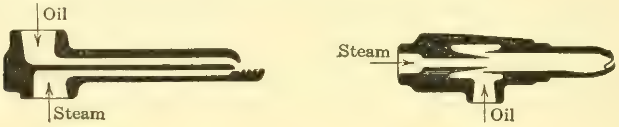
Atomizing burners are the ones commonly used in marine, locomotive, stationary power, and heating installations. The atomization is effected ordinarily by *spraying* produced (1) by *steam jets* (and sometimes by jets of compressed air or gas), or (2) by *mechanical* means.

(**b**) In the **steam-atomizing burner,** the function of the atomizing fluid is to shear the oil into very fine particles and project these into the furnace. Burners of this type may be classified as (1) *outside-mixing* and (2) *inside-mixing.*

One form of **outside-mixing burner** is shown in Fig. 637 (*a*). Such burners are usually arranged to give a *flat flame,* in which case the air for combustion is generally supplied through checker work which forms a portion of the furnace hearth. With this shape of flame and method of air supply, the furnace is limited to but one row of burners across its width. If a flat flame burner is forced, then atomization is not completed, and this results in slower burning, accompanied with excessive smoking, fouling of boiler surface, and lowered efficiency. Oil-fired locomotives commonly use burners of this type. In stationary practice, such burners are usually limited in application to units which operate at moderate rates and without wide variation in loading.

One design of **inside-mixing burner** is shown in Fig. 637 (*b*). This, too, gives a flat flame. Other burners of this type are arranged to give a hollow conical flame to which the air is supplied through controllable registers which are placed around and form a part of the burner casing. The air is induced by the furnace suction and the aspirating effect of the expanding steam jet in the burner nozzle. When large capacity is desired, the air is usually supplied to the register under pressure produced by a forced draft fan. The conical flame is short, bushy and soft, so that air penetrates it rapidly. As more than one row of these burners may be placed across the furnace, almost any reasonable capacity can be obtained from a steam-generating unit, provided the furnace has ample volume.

With steam-atomizing burners, the oil temperature required at the entrance to the burner to give a suitable viscosity for atomization is usually between 150 and 200 deg. fahr. The steam is supplied to the jets at pressures of from 25 to 80 lb. per sq. in. gage; and from 1 to 7 per cent of the steam generated is used for pumping, heating, and atomizing the oil.



(*a*) (*b*)

FIG. 637.—Simple Steam-Jet Oil Burners. (*a*) Outside-Mixing Burner. (*b*) Inside-Mixing Burner.

While burners of this type are simple to manufacture, install and operate, and will burn the oil satisfactorily under normal conditions, they may be objectionable because of (1) the noise they produce; (2) their blowpipe action, which may damage the walls in combustion chambers of unsuitable material, size and proportions; (3) the added moisture in the flue gas; (4) the waste of feedwater due to the steam used for atomizing; and (5) the limitation in maximum capacity obtainable with them in the steam-generating unit. With a single row of flat flame burners a simple boiler cannot be forced ordinarily to a rate of evaporation much above 7 lb. of steam per hr. per sq. ft. of heat-transmitting surface. With burners having air registers and using forced draft, much higher rates can be obtained.

Steam-atomizing burners are extensively used in locomotives and small power plants largely because of their simplicity and low first cost. Air-atomizing burners are used in many heating installations.

(c) In **mechanical-atomizing oil burners,** no auxiliary atomizing fluid is used. There are two main types of these burners: (1) the *rotary burner,* and (2) the *spray-nozzle burner.*

In the **rotary burner,** the oil is fed through a hollow rotating shaft on the end of which is usually a hollow cup which throws the oil from its edges in the form of small particles. Rotary or "centrifugal" burners are used only in small low-pressure installations, and will not be considered further.

The **spray-nozzle burner** is practically the only kind of mechanical burner used with power-plant steam-generating units. In such a burner the oil, under its own pressure of from 30 to 300 lb. per sq. in. gage and with temperature from 100 to 350 deg. fahr. is sprayed from a small nozzle in which, by means of suitable passages, it is usually given a whirling motion before it is suddenly liberated. The oil is discharged from a small orifice concentric with its axis of rotation and issues in the form of a hollow cone. The combustion air is admitted, under either furnace draft or forced draft, through a register surrounding the burner, and may be introduced in such manner as to have whirling motion with respect to the flame. The oil-burning portion of the combination burner shown in Fig. 604 is a typical arrangement of a burner of this type. Modern mechanical-atomizing burners have capacities up to 5000 lb. of oil per hr. per burner; and several rows of them can be installed in the furnace wall if high capacity is required of the unit. Mechanical-atomizing burners are used in most marine installations, partly because the make-up water required for a plant with such burners is less than when steam-atomizing burners are used. Also, the mechanical-atomizing burner is widely used in large central power stations largely due to the belief[2] that this type of burner gives better fuel preparation over a wider range of loads than the steam-jet burner.

(d) With the **methods of regulation** used in the earlier installations of oil burners, the weight of oil delivered by the burner per hour was adjusted by varying the oil pressure, by changing the burner tips, or by operating a different number of burners. But these methods were somewhat unsatisfactory. The rate of oil flow through the burner is substantially proportional to the square root of the oil pressure at the burner; thus, a 50 per cent pressure reduction decreases the flow only about 30 per cent. Furthermore, proper atomization is not obtained with oil pressures below 50 lb. per sq. in. Replacing the burner tips takes time and interrupts operation. Varying the number

[2] Trans. A.S.M.E., Vol. 50, 1928, FSP–50–13, p. 106.

of burners in use in a given furnace results in poor air distribution and uneven use of the furnace volume.

Later methods of regulation are to by-pass or return a governed quantity of oil from the burner to the supply tanks or burner header, or else to have two independent sets of oil feeds to the burner tips, either or both of which may be used. These methods permit of minimum loads of from 15 to 35 per cent of the maximum burner capacity without material change in the nature of the fuel spray.

446. Oil-Fired Furnaces.—(a) The general considerations underlying the design of a furnace in which fuel oil is burned are much the same as when pulverized coal is the fuel, with the principal exception that the ash is not a source of so much trouble. The stages through which oil-furnace design has progressed will indicate what the main considerations are. Early furnaces with steam-jet oil burners were adaptations of hand-fired coal furnaces with the grates bricked over and with steam atomizing burners installed through the front fire doors. The difficulties experienced were *incomplete combustion* and rapid deterioration of the furnace setting and boiler tubes due to *local flame impingement.* To remedy the flame impingement, the burners were shifted to the bridge wall and the spray directed toward the furnace front. This constitutes the "rear-shot" arrangement. With vertical gas passages and boiler tubes inclined upward toward the front of the furnace, there is an increase in volume along the path of flame travel and less impingement if the flame travels toward the front. The incomplete combustion was largely due to insufficient distance of flame travel before striking heat-absorbing surfaces and to improper mixing with air; furnace volumes were so small that combustion rates up to 75,000 B.t.u. per hr. per cu. ft. of furnace volume were common. To give a longer path of flame travel, the grates were removed, the burners lowered, and the tubes raised. Complete combustion was then secured and energy liberation rates of around 30,000 B.t.u. per hr. per cu. ft. of furnace volume were obtained and proved to be satisfactory with refractory walls. Locomotives which use the steam-atomizing oil burners are still equipped with rear-shot furnaces, and they have proved very satisfactory.

The spray from a mechanical-atomizing burner does not have as great a penetration as that from the steam-spray burner, so deterioration of furnace setting or boiler tubes from local flame impingement is not so serious a problem when this type of burner is used. In fact, it is claimed [3] that the tip of a mechanical burner atomizing from 600 to

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[3] "The Burning of Liquid Fuel," by E. H. Peabody, Trans. A.S.M.E., 1928, FSP.-50-13, p. 108.

800 pounds of oil per hour may be placed within 5 feet of a boiler tube without causing trouble, provided the tube is clean and allows good circulation of the water and steam.

Above the ignition temperature, the rate of chemical reaction of the combustibles with oxygen is so fast that the rate of energy liberation within the furnace depends entirely on the rate at which oxygen is brought into contact with the combustibles; or, in other words, it depends on the rate of mixing. With a mechanical burner good mixing is possible close to the burner tip, and the furnace volume may be effectively utilized. Thus, by securing intensive mixing, the furnace volume necessary for complete combustion with a given rate of energy liberation has gradually been reduced until energy-liberation rates up to 60,000 B.t.u. per hr. per cu. ft. of furnace volume seem to be allowable, even with refractory walls.

(**b**) The arrangement of furnace for **supplying air** for the oil-firing depends largely upon the type of burner used. In small stationary plants using the *flat-spray* steam-atomizing burner, natural draft is commonly employed and the air is supplied through brick checkerwork in the furnace floor. The air openings in the checkerwork are then designed to distribute the air approximately in the shape of the flame, and the burners are usually installed less than a foot above the floor. With the steam-atomizing burners used in locomotives, either vertical or horizontal draft may be employed. A flash wall is built up almost to the level of the fire door about 6 feet from the burner. With vertical draft, the air is admitted through a flash hole just in front of this wall; the vertical stream of incoming air then meets the horizontal oil spray and turns this spray upward just in front of the hot flash wall. With horizontal draft, the air is admitted through the fire door and flows horizontally to meet and mix with the oil spray traveling in the opposite direction. With the mechanical or steam-jet burners giving a conical flame the air is supplied through air registers surrounding the burner and no other provision for supplying air need be made in the furnace design.

(**c**) The oil-fired furnace has been built with solid refractory walls, with air-cooled walls, and with water-cooled walls. The two principal causes of failure of *refractory walls* in oil-fired furnaces are spalling and slagging. Spalling, or the breaking or cracking of bricks to expose fresh surface, is the major cause of failure and may be influenced by slagging. Since the mineral ash content of fuel oil is very low, the slagging of bricks in oil-fired furnaces should not be serious unless the temperature is high enough to cause actual fusion of the ash with the bricks. The temperatures of the refractory linings of oil-fired furnaces are usually reputed to be very high—around 3000 deg. fahr. Actual measurements of wall temperatures were made at the plant of the Blackstone Valley Gas and Electric Co. of Pawtucket, R. I. In the furnace tested, there were four mechanical atomizing oil burners and the draft was forced. The furnace walls were solid with no air or water cooling, and the bridge wall was made of firebrick. The highest refractory temperature found was in a brick in the side wall about halfway between the front wall and bridge wall on a horizontal level with the lower pair of burners, near the center of the area of flame impingment. With an energy liberation rate of 62,700 B.t.u. per hr. per cu. ft. of furnace volume, the temperature of this brick reached a maximum of 2840 deg. fahr.[4] The maximum gas temperature recorded was slightly under this value; perhaps because surface combustion raised the temperature at the refractory. Conclusions drawn from these careful measurements were: (1) that the maximum temperature measured in the oil-furnace gases was not any higher than has been found in the gases of stoker and pulverized coal furnaces; (2) that this temperature was not high enough to cause fusion of the refractory; (3) that the slagging action is therefore due to the fusion of the ash of the oil, which is small in amount; but that this slag action was probably related in some way to the spalling, since it was found to be about the only action in the oil-fired furnace which was of greatly different character from that in the coal-fired furnace, and therefore it is the only action which might account for spalling causing major trouble in oil-fired furnaces and minor trouble in coal-fired furnaces.

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[4] "Refractories Service Conditions in Furnaces Burning Fuel Oil; Progress Report," Trans. A.S.M.E., 1928, FSP.–50–81, p. 318.

CHAPTER XXXV

THE BURNING OF OIL IN HIGH-SPEED ENGINES

447. Introduction.—(a) In Part I of this text the various cycles of internal combustion motors have been analyzed and the types and mechanical features have also been discussed. Now that the general methods of burning various fuels are being treated, the special problems encountered in burning fuel in the high-speed engine need to be considered also.

High-speed engines are required in airplanes, automobiles, trucks, speed boats and other forms of transportation, because high speeds reduce the weight per unit of power and, furthermore, the cost is also reduced when the weight is kept reasonably low, provided large numbers of the same size and type of engine are manufactured.

(b) The **carburetor type** of engine has been developed to a very satisfactory standard today and is the one used for the great majority of automotive engines. The high speed, ease of control, high mean effective pressure, fuel economy, and general reliability of this engine afford remarkable evidence of the excellent development work that has been done, especially within the past twenty years. On the other hand, the carburetor type of engine cannot use heavy oils nor can it employ as high a compression ratio as the engine that injects the fuel into the cylinder at or near the end of the compression stroke. Since the compression ratio is always a factor in the thermal economy of any internal combustion engine, regardless of what cycle is employed, engineers naturally try to develop the high-speed engine having sufficient compression to produce auto-ignition of the fuel very soon after its injection. If this type of engine is to run at high speed many new problems regarding the injection of the fuel and its combustion are involved, because the time available for these processes becomes so very short.

(c) The *name* that should be given to the internal combustion **engine using auto-ignition** of the fuel sprayed into the cylinder by means of a high-pressure pump is somewhat debatable at the present time. Such terms as "*solid-injection*," "*compression-ignition*" (C.I.) and "*Diesel*" are applied by various engineers. The objections to designating such an engine as a "Diesel"[1] are: (1) Auto-ignition of oil fuel injected into a hot chamber by a pump at the *end* of the compression stroke was first used in an engine developed by Ackroyd-Stuart in 1890, several years ahead of Diesel's engine. (2) Compressed air was the only successful means of fuel-injection developed by Diesel. (3) The cycle used in high-speed injection engines is the dual-combustion one[2] instead of the Diesel. (4) Confusion is caused by employing the term "Diesel" to cover a much wider field than appears justifiable.

The modern high-speed oil engines are largely employing "airless injection" of the fuel, first used by Ackroyd-Stuart. Furthermore, many of the late engines employ an auxiliary bulb or chamber into which the fuel is injected, as was done in the Ackroyd-Stuart engine.[3] Thus, there seems to be small justification for using the term "Diesel" to apply to all modern oil-injection engines.

(**d**) When an oil engine employs **air injection** of the fuel, as in the Diesel, the cost and weight of the air compressor become very objectionable if the engine is to be used in automotive work or in small motor boats. Thus the present tendency in this field is almost entirely toward the development of the airless injection type; for this reason and also because the method of air injection has been presented in Part I, this chapter will be chiefly concerned with airless injection and its difficulties when high speeds are attempted. High speeds, in this sense, generally mean those above 700 r.p.m.; one engine of this type has been built to run as high as 3000 r.p.m., but 1200 to 1800 are much more usual.

448. Fuel Factors.—(**a**) The general properties of fuels have been given in Chapter XXVII; but some factors that are of special importance when the fuel is injected into the high-speed engine require further discussion. At the outset the point needs to be made that high-speed oil engines require oils that have undergone *some refinement,* hence this type of engine is not able to utilize a range of fuels as wide as the slow-speed Diesel, yet the range is much wider than that of the carburetor engine.

The properties of a fuel that will be briefly considered in relation to the high-speed oil engine are: *ignition temperature, vapor pressure, compressibility, specific gravity* and *viscosity.*

(**b**) The **ignition temperature** of a fuel to be used in an engine is the lowest temperature of compressed air that will ignite this fuel when

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[1] Also see The Engineer, Sept. 5, 1930, p. 256, and Mech. Eng., Nov. 1930, p. 270.

[2] See Sect. **133**, p. 185, Part I.

[3] "Development of the High-Speed Oil Engine in England," by Alan E. L. Chorlton, Mech. Eng., June, 1931.

sprayed into this air. The values as obtained by Tausz and Schulte[4] are shown in Fig. 638 for several oils. Note that there is a wide variation in the ignition temperatures of the fuels shown and that the ignition temperature of all of them is reduced by increasing the air pressure, but especially noticeable is this for pressures up to 15 or 20 atmospheres. Until the results of these investigators were known the general assumption had been that the ignition temperature of an oil was independent of pressure. These results are considered reliable because every precaution was taken to secure uniform temperature of the air in the ignition chamber and to prevent catalytic influences. The chamber was lined on the inside with asbestos and the ignition points were determined by injection when the temperature was falling rather than when the chamber was being heated by the application of an external flame. Tausz and Schulte state that: "Ignition depends on the chemical composition of the fuel and the preceding chemical changes, which differ for different pressures and temperatures. Physical conditions, such as fineness of division and diffusion, have no immediate effect on the ignition temperature but affect only the completeness and rapidity of the combustion."

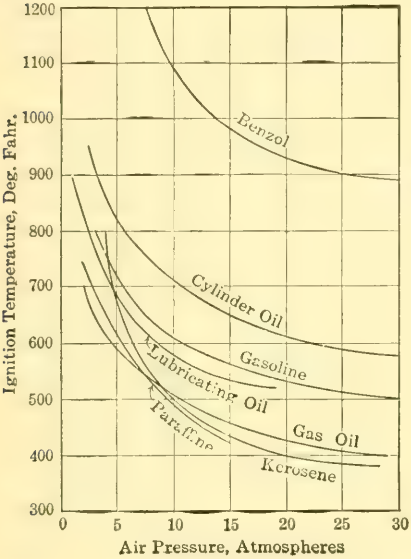

FIG. 638.

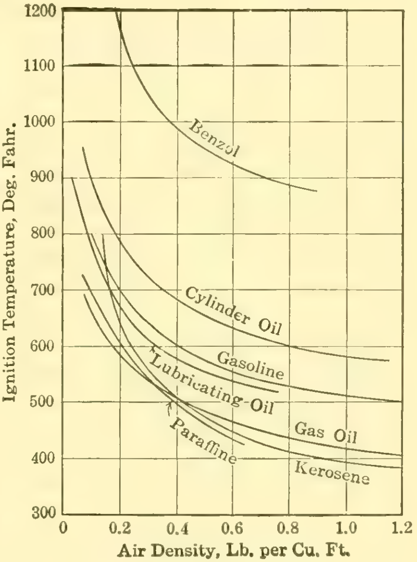

FIG. 639.

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[4] Z.V.D.I., May 31, 1924, p. 574; Translated in Tech. Memo. N.A.C.A., No. 299, 1925.

The later work of Neumann[5] shows that heat transmission from the air to the fuel particles depends upon the *density* of the air, and he therefore believes that the ignition temperature depends primarily on the air density rather than upon its pressure. By calculating the air density for the various pressures and temperatures involved in Fig. 638 the ignition temperatures of these fuels may be plotted against density, as shown in Fig. 639. These curves are quite similar in the two figures and show that for the same density there is a wide range in the ignition temperatures of the different oils.

The ignition temperature of the liquid was found to be lower than that of its vapor for a large number of oils tested by Wollers and Ehmcke. Their results[6] are given in Table LV.

TABLE LV

IGNITION TEMPERATURES OF OIL GASES AND LIQUIDS IN AN OXYGEN STREAM AT 1 ATMOSPHERE

	Gas		Liquid	
	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.
Paraffin oil	614 to 655	1137 to 1211	240	464
Light oil from primary tar	615 to 651	1139 to 1204	326	587
Coal-tar oil	645	1193	445	833
Vertical-oven tar	635 to 661	1175 to 1222	468	874

(c) The **vapor pressures** of liquid fuels at different temperatures need to be considered when fuel is intended for use in any kind of an internal combustion engine. Thus, in the carburetor type of engine, if the fuel becomes too hot the fuel pump or the intake manifold may become nearly filled with vapor and thus the system becomes "*vapor locked.*" With an injection system in which the oil is heated the same trouble may also develop, or the presence of a small amount of vapor in the injection lines may afford sufficient elasticity to prevent the correct delivery of the liquid to the injection nozzle. Whether the vapor pressure is an important element in the ignition lag may be considered somewhat debatable, as the question of how much vaporization precedes ignition is involved; this point is discussed in Sect. **449** (**c**).

The wide range in vapor pressures for various fuels and temperatures is shown in Fig. 640 as found by Joachim and Rothrock.[7] Note that curves *A* and *B* indicate a very small vapor pressure until a temperature of 700 deg. fahr. or more is reached and then the curves turn up rapidly.

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[5] "Experiments on Self-Ignition of Liquid Fuels," Z.V.D.I., Aug. 7, 1926. Translated in N.A.C.A. Tech. Memo., No. 391.

[6] See Mech. Eng., November, 1927, p. 1233.

(d) The **compressibility** of oils is of considerable importance when dealing with high pressures, say from 4000 to 8000 lb. per sq. in., such as those used in airless injection systems. The results of Alexander[8] indicate that for pressures up to about 6000 lb. per sq. in. the volume of fuel oil is decreased 1 per cent for each increase in pressure of 2800 lb. per sq. in. Different oils will vary somewhat and the compressibility increases with the temperature, but only a very little information regarding this point is now available for various oils.

(e) The **specific gravity** of an oil affects the angle and depth of penetration of a spray from an injection nozzle; but the experience of many engineers indicates that the range of gravity that may be used with a particular injection system is large. There has as yet been no general agreement on what constitutes the limits of the gravity that should be used in the different types of engines; but specific gravities from 0.84 to 0.91 are given by Nixon[9] as well within the limits. He further states that the fuel recommended is "*gas oil*" which distills after paraffin (i.e., from 390 to 660 deg. fahr.).

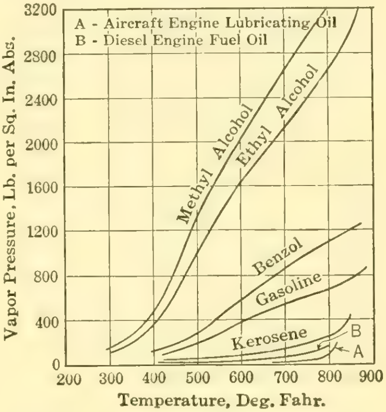

FIG. 640.—Vapor Pressures of Oil Fuels.

(f) The **viscosity** of the fuel used affects its flow through the pumps, distributing lines, and nozzles. Oils of low viscosity generally give a cleaner combustion chamber and smoother operation than do those of high viscosity. Since viscosity is a measure of how rapidly the oil can be atomized, it should probably take precedence over specific gravity in the specifications for fuel to be used in light high-speed engines. For this purpose the viscosity at 100 deg. fahr., as determined by a Saybolt Universal, should probably be between 10 and 45 sec.

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[7] See N.A.C.A. Report No. 321, 1928.

[8] See The Automobile Engineer, June, 1931, p. 234; also Trans. Inst. Marine Engineers, 1927–28, p. 366.

[9] "The Compression-Ignition Automobile Engine," The Automobile Engineer, May, 1931.

(g) **Filtering the fuel** to be used in a high-speed oil engine seems to be the only sure way to remove the very fine particles of dust that are constantly floating in the air and thus getting into an oil that may have been clean when it left the refinery. Fine grit from any source causes very serious trouble with the valves, pumps and nozzles of high-speed oil-injection systems.

449. Processes Within the Combustion Chamber.—(a) When oil is sprayed through a nozzle into the combustion chamber of an engine, near the completion of the compression stroke, it mixes more or less perfectly with the hot air within that chamber; ignition takes place after a short "lag"; combustion occurs; work is delivered to the piston; heat is transmitted to the walls of the chamber and thence to the water jacket. All engineers are sure of these things and they know that only a few thousandths of a second are available for the several processes to occur if the engine is a high-speed one; but as the study is pursued further many questions arise that cannot be positively answered by any one at present. Thus, whether the liquid ignites and burns from the liquid phase, or whether considerable vaporization is required before ignition, or whether something else is involved in the ignition, as outlined in Sect. **444,** still remain very perplexing questions; but the evidence at present seems to indicate that vaporization is of minor importance. This evidence comes largely from the study of the phenomenon called "ignition lag."

(b) The **ignition lag or delay** is the time from the first entrance of the fuel spray into the combustion chamber until ignition takes place. If the air is at a temperature from 50 to 100 deg. fahr. above the ignition temperature of the fuel the ignition lag is very short; but when the air is at the minimum temperature that will produce ignition, and the oil is cold, the lag may be very great, say as much as 3.5 seconds, as found by Hawkes[10] in one experiment as early as 1920. Neumann[11] showed that the ignition delay could be very much reduced by the *turbulence* of the heated air into which the fuel is sprayed, and that the higher the temperature of this air above the ignition temperature of the fuel the shorter becomes this delay. In fact his experiments showed that this lag is influenced by those factors that affect the rate of heat transmission from the hot air to the minute liquid particles. Thus, in addition to turbulence and air temperature, the lag is affected by the temperature of the fuel, the degree of atomization, the distribution of the spray in the combustion chamber, and the rate of radiation from the hot parts of this chamber.

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[10] See Engineering, Vol. 110, 1920, p. 787.

[11] See Z.V.D.I., Aug. 7, 1926; also N.A.C.A. Tech. Memo., No. 391.

The magnitude of the ignition lag in modern high-speed oil engines is of the order of 0.001 to 0.002 sec., according to Joachim.[12] This seems, at first thought, to be such an extremely short time that one need not consider it; but when an engine is running 1800 r.p.m. an interval of 0.002 sec. will permit the crankshaft to turn 21.6 deg., which is a movement easily visualized by any one, and thus these minute intervals of time become very significant in designing the injection system.

The ignition lag obtained by Davies and Griffen[13] with a single-cylinder engine having an auxiliary air chamber is given in Table LVI, both in crank angle and in time, for a wide range of speeds.

This table shows that for this engine the ignition lag involved nearly the same crank movement over a wide range of speeds; the same ignition lag in degrees at all speeds indicates that this engine is so constructed that the rate of heat transfer to the liquid fuel increases directly with the speed.

TABLE LVI

Speed, r.p.m.	750	1000	1250	1500	1750	2000
Ignition lag:						
Crank angle, in deg.	14	14	11	13	11	11
Time, in sec.	0.0032	0.0024	0.0015	0.0015	0.0011	0.0009

The ignition lag involves so many factors that the complete correlation of all of them will probably require many more years of painstaking research work. The "*ignition lag*" should not be confused with the term "*injection lag*" explained in Sect. **450** (**d**).

(**c**) The **theories regarding ignition** of oil, as outlined in Sect. **444,** may now be examined somewhat critically because the evidence obtained from the study of the ignition lag in oil engines, from the ignition temperature of oils and their vapors, and from the operation of engines with various oils, has been very illuminating and in many cases rather startling. Thus, in Table LV the ignition temperatures of a number of liquid oils were shown to be less than those of their own vapors; these facts are hard to reconcile with the vaporization theory of ignition. Furthermore, Neumann,[14] using the laws of heat transmission, has shown that for his experiments not more than 5 per cent of the liquid can be vaporized in the time available with a long ignition delay and with great turbulence of the air; he also found that with a short ignition delay (0.018 sec.) the evaporation was only 0.1 per cent. He concluded therefore "that no preliminary evaporation of the fuel is necessary for producing the ignition."

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[12] See "Combustion in High-Speed Oil Engines," by William F. Joachim, Trans. A.S.M.E., 1929, OGP–51–11, p. 99.

[13] See "Injection, Ignition and Combustion in High-Speed Heavy-Oil Engines"; The Automobile Engineer, June, 1931, p. 233.

[14] Z.V.D.I., Aug. 7, 1926; translated in Tech. Memo. N.A.C.A., No. 391, 1926.

If evaporation were a controlling factor in ignition then naturally one would expect that those fuels which evaporate most readily would give the best results in an engine; but this expectation is not supported by their actual use in an engine. For example, benzene, C_6H_6, has a flash point at 10 deg. fahr. and boils at 176 deg. fahr.; but this fuel is very difficult to burn in an oil engine. On the other hand some of the best oils for such use do not begin to distill until a temperature of 400 deg. fahr. or higher is reached.

Neumann also calculated the rate of heat transmission from the compressed air to the fuel drops, on the assumption that these drops were heated to their known ignition temperature during the time of the ignition lag solely by heat transmitted from the air. These calculations gave him mean coefficients of heat transfer that were of the order of 100 times larger than the normal values, thus supporting the theory that some form of *chemical action* must assist heat transfer during the ignition lag. Further support of the *activation* theory comes from Callendar's experiments[15] on the effects of peroxides in nuclear drops. When studying combustion in tubes placed side by side in a furnace, one tube traversed by a very fine spray and the other by a completely vaporized mixture, he found that "peroxides were found in great profusion in the liquid drops while the vaporized mixture showed very little effect. The peroxides present in the drops caused ignition by intermittent flashes at a much lower temperature than that required to ignite the vaporized mixture, which burnt with a steady flame when the temperature was sufficiently raised." He also states that "each flash is presumably started by the ignition of the peroxides accumulated in the drops, and is followed by a brief period of quiescence during which the concentration of the peroxides increases till it suffices to start another flash."

Regardless of what additional facts may be determined by future experiments, enough is now known to show that the ignition and combustion of oil in an engine constitute a very complex problem and that vaporization certainly should not be considered as a self-evident stage preceding ignition.

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[15] "Dopes and Detonation," Engineering, Feb. 4, 1927, p. 147.

(**d**) **Detonation** is not so serious in the high-speed oil engines as in the carburetor type[16] having a much lower compression ratio. However, as the former type is now being developed to give higher mean effective pressures and higher speeds, detonation has already given evidence of becoming troublesome in some engines. If the fuel is heated too much before injection, and if the injection begins too early, the trouble is aggravated; but no one knows the general relation between compression pressure, method of injection, rate of injection, shape of combustion chamber, engine speed and detonation.[17]

450. The Fuel-Injection System.—(**a**) The airless injection system consists of high-pressure pumps, distributing lines, and injection nozzles. Since the time of injection, the degree of atomization, and the metering of the fuel used per cycle are dependent upon the design and maintenance of this system, its importance is apparent. The extremely short time of the injection period, the very small quantities of fuel to be metered, and the high injection pressure, say 4000 to 8000 lb. per sq. in., all combine to make unusual precision in manufacture of the pumps and nozzles of the utmost importance. As the speed of the engine increases and the size of the cylinders becomes smaller, the amount of fuel to be injected per cycle becomes extremely minute. For example, a six-cylinder, four-stroke cycle engine developing 75 h.p. at 1200 r.p.m. and using 0.45 lb. of fuel per b.h.p.-hr. would require only 0.000156 lb. of oil per injection; this means approximately 0.005 cu. in. of oil to be delivered to each cylinder at intervals of time only 0.1 sec. apart, and the actual length of an injection period may be only 36 degrees of crank movement, or 1/200 sec.

(**b**) The **two systems of airless injection** of fuel are often designated as (1) the "*common rail*" and (2) the "*jerk pump*" or "*individual pump*" systems.

In the **common-rail system** a pump (usually with three plungers) delivers oil at substantially constant pressure to a *common rail* or *line* from which the oil is carried by small tubes to the injectors in the several cylinders of the engine, as indicated in Fig. 641. With this system the sole function of the pump is to maintain the oil pressure in the *common rail*, or reservoir, at the desired value; hence the metering of the oil and the timing of the injection period are not affected by the pump. If the engine uses less oil than the pump is delivering, the relief valve allows the excess to return to the intake of the pump.

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[16] Discussed in Sect. **381**.

[17] Some information regarding detonation in oil engines is given by Griffith and Vincent in Trans. A.S.M.E., 1930, AER–52–5, p. 41.

The hand pump is used only to bring the oil pressure up to the desired value to permit injection when starting.

In the **individual pump system** a separate pump is provided for each cylinder of the engine. These pumps are usually cam-driven with a spring return, thus making a variable length of stroke easy to attain; and the cam is so designed that the pump plunger moves at high speed during most of its discharge stroke. The amount of oil delivered by the pump per stroke is regulated by various means such as by-passing, throttling the intake, or varying the length of stroke. The time of injection is determined by closing the by-pass port that is open during a considerable part of the delivery stroke. Consequently, this type of pump is often referred to as the *timed pump,* whereas the common-rail system uses a *non-timed pump.*

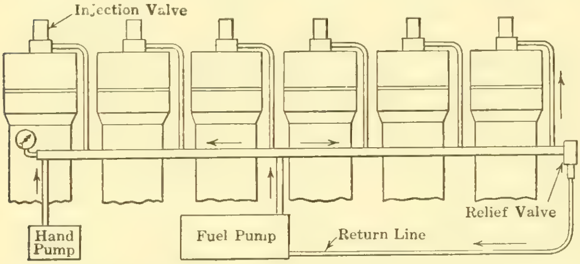

FIG. 641.—Diagrammatic Sketch of a Common-Rail Fuel System.

The difficulties of obtaining equal distribution of fuel to the cylinders of an engine operating at various speeds and loads are very great in any fuel-injecting system; but they appear to be greater with the common-rail system, and the preference, at present, therefore seems to be for the individual-pump system.

The injection pump must handle such small volumes of fuel under such high pressures that *packing* cannot be successfully used, as is done with low-speed engines. Instead, the plungers and cylinders are hardened and ground to such a close fit that leakage is very small even with a short plunger. One company specializing in this class of work reports that a difference of four millionths of an inch in a plunger 0.876 in. in diameter would cause its rejection.[18]

(c) The **injection nozzles** are designed to form the fuel into a very fine spray and distribute this spray throughout a definite portion of the combustion chamber. These chambers are made in a variety of shapes, as shown in Sect. **451**, and the nozzle requirements are naturally somewhat different for various cases. These nozzles are often called *injection valves,* also *spray valves* or nozzles, and sometimes simply *injectors.* They are divided into two general classes, designated as the *open* and *closed* types.

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[18] See Trans. A.S.M.E., OGP-50-7, 1928, p. 1.

The **open nozzle** has no means provided within it to stop the flow of oil into the cylinder, but it should contain a check valve, as shown in Fig. 642, to prevent the entrance of gases from the cylinder; in other words, it is always open to the cylinder except when the check valve is closed, and the flow of oil through an open nozzle must be therefore entirely controlled by the pump, or by special mechanically operated timing valves in a common-rail system.

The **closed nozzle** contains a spring-loaded valve that may be opened either *mechanically* or *automatically*, the latter by the fuel pressure when sufficiently high. The principle of operating a closed nozzle automatically may be understood from Fig. 643. The spring is adjusted so that the desired pressure of oil will just suffice to open the valve. After the valve has started to open, the area on which the oil acts is then increased slightly; hence a pressure wave coming through the pipe line from the pump or from a common-rail system will open the valve very quickly. When the closed automatic nozzle is used with the common-rail system, one or two mechanically operated valves are essential to control the time during which the high-pressure oil is acting on the nozzle. If only one mechanically operated valve is employed, the injection nozzle will open when this valve admits the high-pressure oil and will remain open after the valve has closed until the spring pressure overcomes the oil pressure remaining in the tube. Sometimes a second mechanically operated valve is employed to release the oil pressure on the nozzle, which is then sure to have a sharp cut-off.

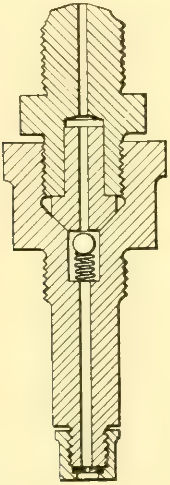

Fig. 642.—An Open Nozzle.

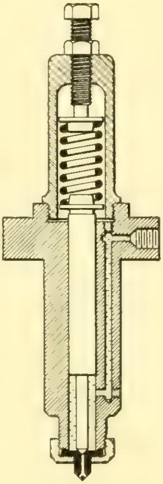

Fig. 643.—A Closed Nozzle.

The great advantage of the closed type of nozzle is the prevention of *dribbling* or *dripping* of the oil after the injection is supposed to stop. Thus a more definite control of the injection period is obtained than is possible with the open type of nozzle which permits dribbling as long as the pressure in the oil line is greater than that in the combustion chamber.

The closed nozzle also has the advantage of giving more nearly a *uniform spray* during the entire injection period, because the oil pressure on the nozzle is high during this entire period. On the other hand, the open nozzle generally has its flow controlled entirely by the pump, and the start and finish of each injection are at lower pressure and the resultant spray at these times may be poor. Furthermore, when the engine is running at low speeds the spray from the open nozzle is much coarser than it is at full speed.

The open nozzle has the advantage of extreme simplicity and generally greater freedom from trouble due to air in the pipe line than the closed nozzle.

The closed nozzle must be very carefully designed and manufactured[19]; it must also be maintained in excellent condition, otherwise the high-pressure oil will surely leak past the valve, and this will soon prove to be a very serious trouble.

Recent investigations[20] show that when the rate of pressure rise at the orifice is high, open nozzles give spray-tip velocities and penetrations which compare favorably with those of closed nozzles.

(**d**) The **injection lag** is the time required for the start of the spray from an automatic injection nozzle after the opening of the timing valve in a common-rail system; and in an individual-pump system this lag would be measured after the closing of the port, or by-pass valve, at the pump.

Tests recently reported by Rothrock[21] show that the injection lag in the common-rail system is decreased by using high injection pressures and short tubes, but is not affected by variations in tube bores ranging from $\frac{1}{16}$ to $\frac{5}{32}$ in. He also found that the initial pressure in the tube before opening the timing valve does not affect the lag when the injection pressure is considerably above that required to open the valve. Since this initial pressure and the tube bore do not affect the lag, he concludes that "the valve is opened by a *pressure wave* and not by a pressure caused by compression of the oil in the tube from the inflow through the timing valve." His results for an initial pressure of 1000 lb. per sq. in., a valve opening pressure of 4000 lb. per sq. in., four different tube lengths and various injection pressures are shown in Fig. 644. The lag is plotted in time units rather than in terms of crank movement because, if a constant pressure is maintained on a common-rail system regardless of engine speed and if the timing valve opens wide quickly, the injection lag in such a system is independent of engine speed. Note from the figure that for this design of system an injection pressure of 9000 lb. per sq. in. and a 10-in. tube gives an injection lag of only $\frac{1}{1000}$ sec., but that reducing the injection pressure to 4000 and increasing the tube length to 21 in. will give an injection lag twice as long.

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[19] The extreme precision required in this work and also in making injection pumps may be appreciated somewhat by reading the article by C. R. Alden, Trans. A.S.M.E., OGP-50-7, 1928.

[20] See "Some Characteristics of Fuel Sprays from Open Nozzles," by A. M. Rothrock and D. W. Lee, N.A.C.A. Tech. Note, No. 356, November, 1930.

[21] See "Injection Lags in a Common-Rail Fuel Injection System," N.A.C.A. Tech. Note, No. 332, February, 1930.

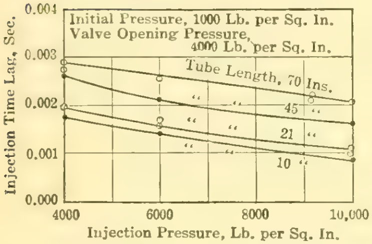

FIG. 644.—Injection Lag in a Common-Rail System.

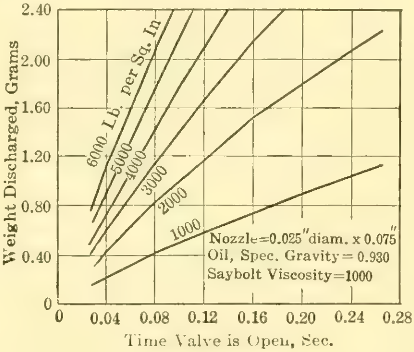

FIG. 645.—Discharge from an Injection Nozzle.

(e) The **weight of oil discharged** from an injection nozzle is determined primarily by the size of aperture, the time the nozzle is open, and the injection pressure. The relation between these factors for one nozzle and one oil is given by the curves of De Juhasz [22] in Fig. 645. The pressure indicated on each curve is the injection pressure. Note that the curves are not quite straight lines because the drop in pressure at the nozzle becomes greater as the period of injection is increased.

Observe that, according to these test results, the ratio of the weights discharged in any given time for two specified pressures is not proportional to the square root of the ratio of these pressures, but to a much smaller root, such as 1.2 or less. On the other hand, Professor Schweitzer [23] states that for short orifices with a length up to four times the

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[22] See "Some Results of Oil-Spray Research," Trans. A.S.M.E., OGP-51-9, 1929.

[23] See Trans. A.S.M.E., 1930, OGP-52-15, p. 121.

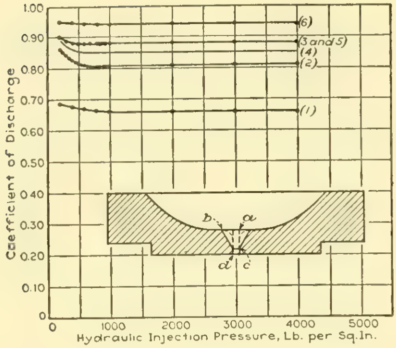

FIG. 646.—Effect of Hydraulic Injection Pressure on Coefficient of Discharge.

(Curve (1) with leading edge *a* sharp; (2) with leading edge *a* rounded to approximately $\frac{1}{64}$-in. radius; (3) leading edge rounded to $\frac{1}{32}$-in. radius; (4) leading edge beveled 60 deg., corners *b*, *c*, and *d* sharp; (5) leading edge beveled 60 deg., corner *b* slightly rounded, *c* and *d* polished; (6) leading edge beveled 60 deg., corners *b*, *c*, and *d* slightly rounded. Orifice diameter 0.015 in.; back pressure atmospheric.)

diameter the discharge increases with the square root of the pressure and is independent of the viscosity. The work of Kemper[24] regarding the coefficients of discharge from small orifices under high-injection pressures also supports the square-root law. His work, as indicated by the results given in Fig. 646, shows that the **coefficient of discharge** is independent of pressure from 1000 to 4000 lb. per sq. in., but is very sensitive to slight changes in the shape of the entering edge of the orifice. These coefficients should be used with the following formula for orifices similar to the ones given in this figure:

$$w = Atdc_d\sqrt{2g \cdot \Delta P/d}\ . \qquad (582)$$

where w = weight in lb. discharged in the time t.
A = area of orifice in sq. ft.
d = density of oil in lb. per cu. ft.
t = time in sec. that the valve is open.
c_d = discharge coefficient of the orifice.
g = acceleration due to gravity in ft. per sec^2. (32.2).
ΔP = Pressure drop through orifice in lb. per sq. ft.

The injection nozzle is open such a very small fraction of a second that the velocity of the spray is much less, near the beginning and end of the period, than the maximum velocity corresponding to the maximum injection pressure. The experimental results given by De Juhasz show that this difference is far greater with high-injection pressures (say 6000 lb. per sq. in.) than with lower ones.

(f) The **spray** from an injection nozzle needs to be finely *atomized* to give satisfactory combustion; but if the drops of oil are too small the *penetration* within the combustion chamber may not be sufficient. If the combustion chamber is designed to give but little turbulence, the spray needs to spread out much more than when there is a high degree of turbulence. The total penetration of a spray depends chiefly upon the density of the gas in the combustion chamber and also upon the size of the orifice, but not very much upon the injection pressure. A high-injection pressure gives a high velocity but also a fine atomization, and these two factors counteract each other very largely as far as the maximum penetration possible is concerned; but the distance penetrated in a *very short time* is much greater with the high-injection pressure, as shown by Fig. 647, obtained from the N.A.C.A. tests [25] with a single orifice 0.015 in. in diameter. The importance of time in this connection is best appreciated by considering, for example, an engine in which the spray penetration desired at the end of 0.002 sec. with the given nozzle is to be 2.25 in. when the chamber pressure is 300 lb. per sq. in.; the curves show that the injection pressure required would be 2000 lb. per sq. in. Now if the same penetration were to be obtained in half the time, or 0.001 sec., i.e., with engine speed doubled, the same set of curves shows that the injection pressure would have to be 6000 lb. per sq. in., or three times as much as before. This indicates that the injection pressure should increase about as the 1.6 power of the engine speed to obtain the same penetration in the time available with a given angular movement of the engine crankshaft. But all things considered the general rule of *increasing the injection pressure about as the square of the engine speed* has been found satisfactory.

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[24] See Trans. A.S.M.E., 1930, OGP-52-5, p. 31.

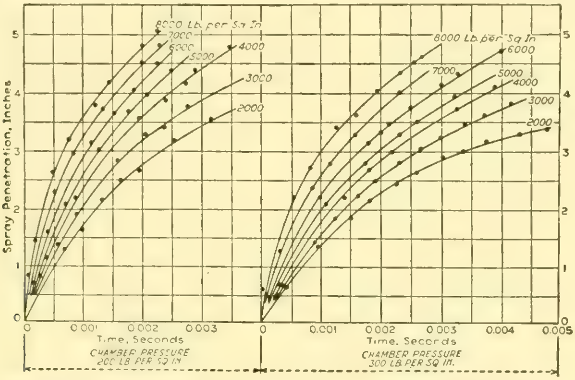

FIG. 647.—N.A.C.A. Curves Showing Effect of Injection Pressure on Penetration.

451. Combustion Chambers.—(a) There are many varieties of combustion chambers in use in high-speed oil engines as made by different manufacturers; certain types are highly favored by some engineers because these types have been found very satisfactory; but many basic truths regarding the best shape of combustion chamber to use with the various injection systems, fuels, and engine speeds are still to be discovered. In all cases the desire is to have the combustion of the fuel completed during the early part of the working stroke in order to give high thermal economy, high mean effective pressures, and to prevent a smoky exhaust. The short time available for injection, ignition and combustion when an engine is running 1000 r.p.m. or more makes the problem a very complex and difficult one.

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[25] See Tech. Report No. 222; also Trans. A.S.M.E., 1930, OGP-52-5, p. 123.

(**b**) The **main types** into which the combustion chambers are usually divided are:

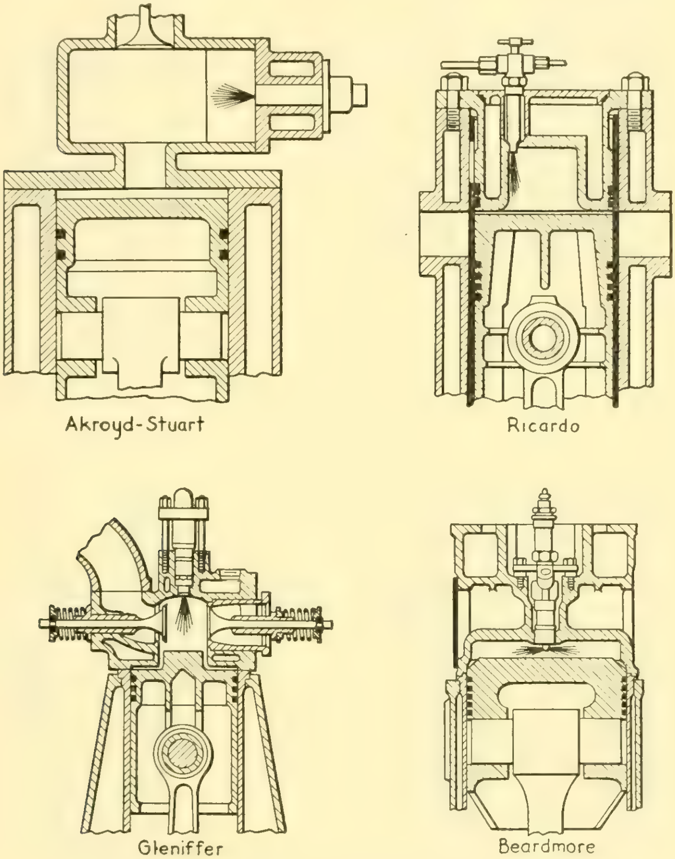
(1) The "**direct injection**" or **single-chamber type**, in which the fuel is injected directly into the combustion chamber, and the required relative motion between the air and fuel to obtain good mixing is assisted (*a*) by the swirl of the air produced by the shape of the entering air ports, and therefore designated as *port swirl*, or (*b*) by the swirl produced by the piston as it completes the compression stroke, and consequently called *piston swirl.*

(2) The **ante-chamber type** in which the main combustion space is connected by a restricted opening to an ante-chamber *into which the fuel is injected* and partially burned, thus creating a difference of pressure in the two chambers that forces a large portion of the mixture into the main chamber; the high velocity produced in the constricted opening between the two chambers produces such good mixing of the hot air and fuel that such engines do not require as fine a spray nor as high a compression pressure as those of type 1.

(3) The **auxiliary air chamber** or "**Acro**" **type** which has an auxiliary air chamber joined to the main chamber by a narrow funnel into which the fuel is injected. This type is considered by some authors as a special case of type 1, and by others as a special case of type 2. Some of the combinations that have been made are hard to classify.

(**c**) The **main advantages and disadvantages** of each type need to be considered briefly. The direct injection type has the advantages of simple combustion chamber, highest mean effective pressures, highest thermal efficiency, applicability to large engines as well as small ones, and ease of starting when cold. The disadvantages of this type are: high maximum cylinder pressures, and very high injection pressures that require great refinement in the design and maintenance of the injection system.

The main advantages of types 2 and 3 are found in their ability to use heavier grades of fuel, lower injection pressure and a somewhat simpler injection system. Their disadvantages are: more complicated cylinder heads, greater loss of heat from the combustion chamber and less satisfactory scavenging. In some cases the loss of heat from the ante-chamber is so great when starting a cold engine that electrical coils are installed in these chambers to overcome this starting difficulty.



Akroyd-Stuart

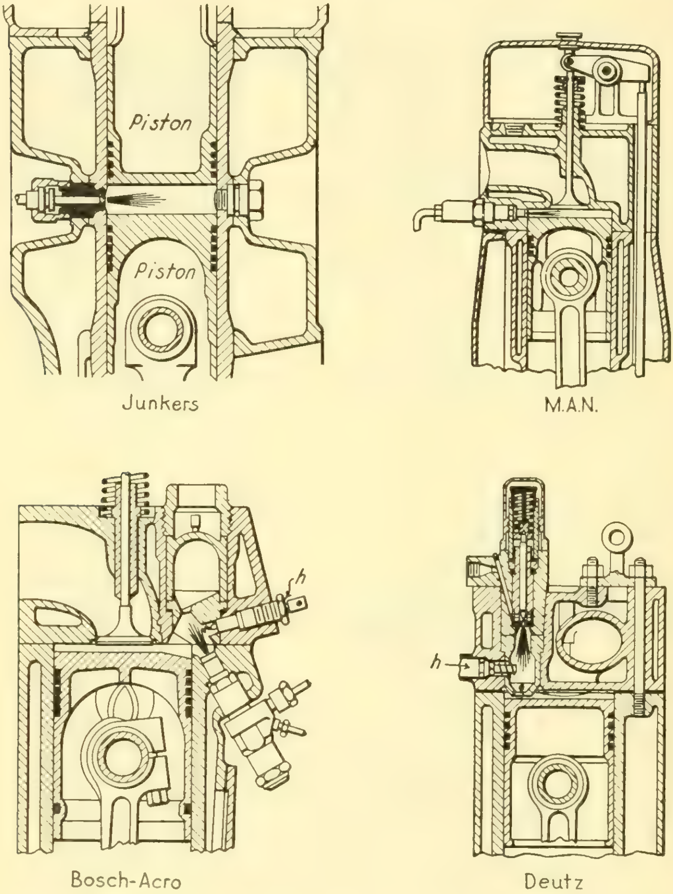
Ricardo

Gleniffer

Beardmore

Fig. 648.—Some British Types of Combustion Chambers.

(**d**) **Some examples** of the various types of combustion chambers are illustrated in Figs. 648, 649 and 650. Of the British combustion chambers shown in Fig. 648, the *Ackroyd-Stuart*, as originally built in 1890, is of special interest as it is clearly the pioneer of the ante-chamber type. The other three examples shown in this figure are of the direct injection type. In the *Ricardo* sleeve-valve engine the combustion chamber is a plain cylinder with a diameter about half that of the main cylinder, and the ports are arranged to give such a violent tangential movement to the air entering the main cylinder that the spray does not need to be so fine as usual with this general type of chamber. The figures showing the *Gleniffer* and the *Beardmore* combustion chambers are self-explanatory. Some of the other well-known British engines are of the other types.



Junkers M.A.N.

Bosch-Acro Deutz

FIG. 649.—Some German Types of Combustion Chambers.

Typical illustrations of some German combustion chambers are given in Fig. 649. The *Junkers* is clearly of type 1 and is unique in that two pistons are used with one combustion chamber. This two-stroke cycle engine obtains a high degree of turbulence by means of guides which direct the air tangentially into the cylinder. The ratio of surface to volume in the combustion chamber is exceptionally small and thus the engine starts easily. The *M.A.N.* (*Maschinenfabrik Augsburg-Nürnberg*) has a somewhat similar combustion chamber, but with the air inlet valve in the center so that the air stream is almost at right angles to the fuel spray.

The *Bosch-Acro* is a modification of the British-Acro having the auxiliary air chamber, and uses an electrical hot-wire heating unit, as shown at *h* in the figure, to assist in starting. The *Deutz* is clearly of the ante-chamber type having only small holes connecting the two chambers. An electrically heated glow plug (*h*) is applied to this engine as shown.

Four of the combustion chambers used in American engines are shown in Fig. 650. The *Cummins* primarily belongs in the classification previously given as type 1; but, certain unique features of the injection system and the combustion chamber of this engine make it really in a class by itself. The piston head contains (as shown) a small air chamber or bottle that delivers a blast of air into the main chamber soon after combustion has begun. During the suction stroke of the engine the metering pump delivers the correct amount of fuel through a check valve into a very small annular space in the tip of the injector. The oil remains in this space long enough to be preheated to a considerable degree before injection into the cylinder. The nozzle and injection pump are combined so that the tip of the plunger may be forced nearly to the end of the nozzle. This action delivers the oil spray into the cylinder at such a rate that an excessively rapid rise in pressure is avoided.[26]

The *Hill* is an example of the ante-chamber type of engine using a relatively large single orifice in the injection nozzle and a low injection pressure (about 1500 lb. per sq. in.)

The *De La Vergne* uses a five-hole nozzle centrally located in the head. The air enters the chamber tangentially through the dual inlet valves, only one of which is shown. The spray is directed close to the head of the piston but does not touch it.

The *Packard*[27] is clearly of type 1; it uses very high air velocity produced in the combustion chamber by giving the inlet port leading to the valve the form of a flattened venturi tangential to the cylinder bore. The piston is recessed on one side to give valve clearance and to assist in promoting air turbulence. The one valve serves to admit air and also to exhaust the burnt gases. The nozzle is an open one connected to an individual pump fastened to the side of the cylinder. A glow plug, not shown, is used to assist in starting in cold weather.

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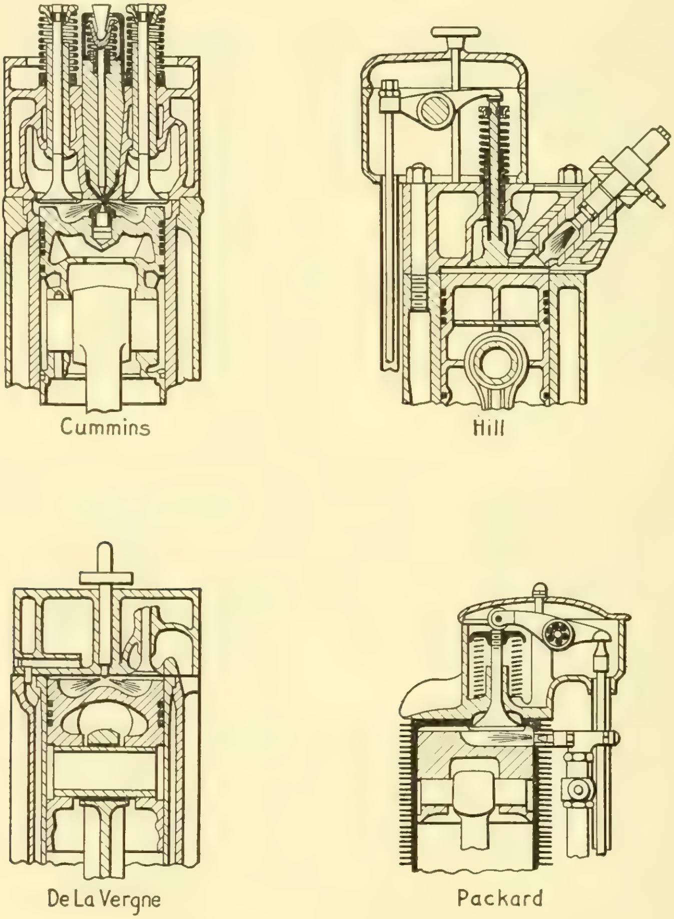
[26] For a complete description of this injection system see S.A.E. Journal, September, 1930, p. 285.

[27] For a complete description of this airplane engine see S.A.E. Journal, April, 1930, p. 431.



Cummins

Hill

De La Vergne

Packard

FIG. 650.—Some American Types of Combustion Chambers.

452. Performance Data.—(a) *Different makes and classes* of high-speed oil engines naturally cannot be expected to give identical performances. However, as previously mentioned, the direct-injection type is usually superior to the others as to thermal economy and brake mean effective pressure. This general truth is brought out clearly by the curves in Fig. 651, as given by Vincent.[28] Note how the curve of specific fuel consumption of the direct-injection type lies below that of the other type except for piston speeds less than about 1000 ft. per min.

(b) The **effect of load** on the performance of a high-speed oil engine is shown by the curves in Fig. 652, as given by Kemper,[29] for an experimental engine having the main cylinder directly connected to a pear-shaped bulb through an orifice $\frac{9}{16}$ in. in diameter; the fuel was injected along the center line of this bulb through a single orifice 0.05 in. in diameter. The curve of indicated m.e.p. is seen to rise to the high value of 140 lb. per sq. in.; but the corresponding brake m.e.p. is only about 70 per cent of this value over a considerable range of loads. To obtain an i.m.e.p. of 140 with a maximum pressure of only 750 lb. per sq. in. is an unusual performance for this type of engine.

The condition of the exit gases, measured within four inches of the exhaust valve, are indicated in the figure; it should be noted that the maximum b.m.e.p. obtained with a clear exhaust is 91 lb. per sq. in. and the corresponding fuel consumption 0.52 lb. per b.hp.-hr.; the minimum fuel consumption (0.46) is obtained at about 60 per cent of full load.

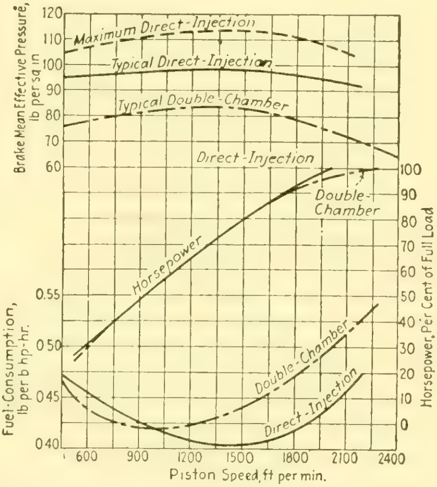


FIG. 651.—Typical Performance Curves of High-Speed Oil Engines.

(c) Various kinds of **indicator diagrams** of an oil-injection engine may yield considerable information regarding the engine's performance; but the **pressure-volume diagram** is not very satisfactory for studying the timing of events, because the whole period of injection, ignition and combustion take place when the piston is so near the end of its stroke that this part of the diagram is badly crowded together. An illustration of the pressure-volume diagram from a high-speed oil engine is shown by the diagram in Fig. 653, from Kemper.[29] The piston displacements corresponding to various crank angles have been added to the figure as an aid in determining the timing.

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[28] See "Status of Compression-Ignition Engine Research," S.A.E. Journal, June, 1931, p. 642.

[29] See Trans. A.S.M.E., OGP–52–5, 1930, p. 33.

To show in less crowded juxtaposition the events occurring near the top (dead) center (T.C.) a **pressure-time diagram**, such as that shown in Fig. 654, may be obtained, as by means of certain types of indicators,[30] and then the timing of these events will be more clearly represented. The abscissas on such diagrams may represent either crank degrees or time as shown.

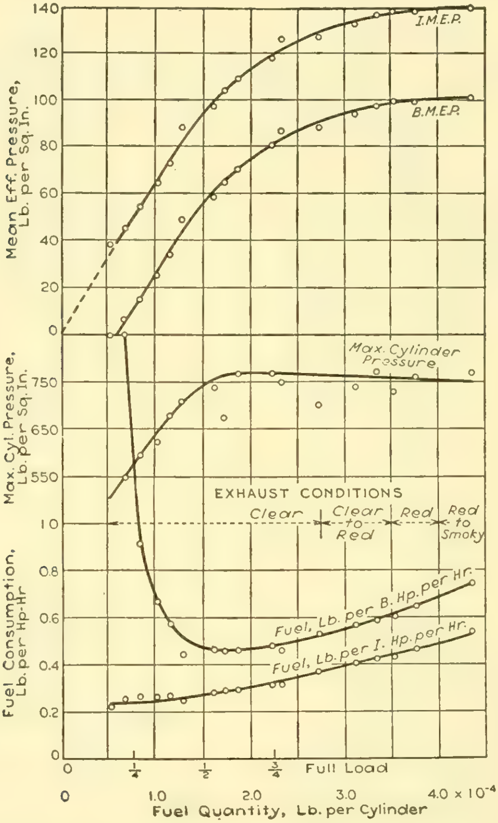

FIG. 652.—Effect of Load on Performance of High-Speed Oil Engine.
Bore, 5 in.; stroke, 7 in.; speed, 1500 r.p.m.

The combustion line may also be studied from an **offset or displaced diagram,** which is obtained with the pressure-volume type of indicator by displacing the reducing motion a convenient amount, 90 degrees being usually preferred.

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[30] For description, see "Experimental Mech. Eng." by Diederichs and Andrae, Chap. 7.

(d) **Some data** ragarding modern high-speed oil-injection engines are given in Table LVII, which lists a few of the various makes in this country and in Europe. The list includes only engines designed to operate at 1000 r.p.m. or more, and is not intended to be a complete one even for such a class. There are many larger high-speed, airless-injection oil engines made to operate at speeds from 500 to 1000 r.p.m. The values given in this table have been obtained largely from the more comprehensive tables of Davies and Giffin,[31] Ruehl,[32] and Nixon.[33] The results already attained, such as those shown by this table, are due to an enormous amount of development work that has been going on for many years; and such work is still being pursued vigorously.

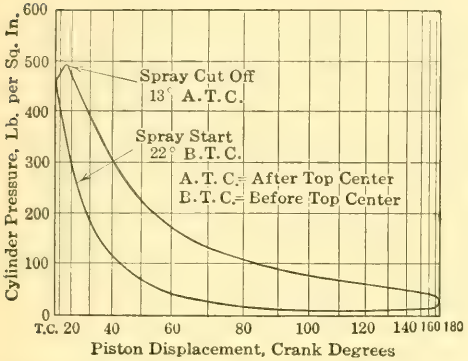

FIG. 653.—Indicator Diagram for High Rate of Injection.

Bore, 5 in.; stroke, 7 in.; comp. ratio, 13.5; speed, 1500 r.p.m.; b.h.p., 22.8

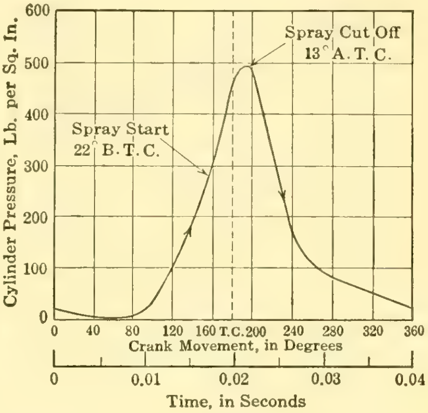

FIG. 654.—Pressure-Time Diagram for FIG 653.

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[31] Engineering, October 24, 1930, p. 532.

[32] S.A.E. Journal, February, 1931, p. 167, also May, 1931, p. 564.

[33] The Automobile Engineer, May, 1931, p. 196.

TABLE LVII

High-Speed Oil-Injection Engines

Name	No. of Cyl.	Bore, Ins.	Stroke, Ins.	R.P.M.	B.H.P.	B.M.E.P.	Comp. Press., Lb. per Sq. In.	Max. Press., Lb. per Sq. In.	Fuel Inject. Press., Lb. per Sq. In.	Inject. Nozzle*	Type of Comb. Chamber†	Specific Fuel Consumption, Lb. per B.H.P.-Hr.	Specific Weight, Lb. per B.H.P.
British Engines:													
A. E. C. Acro	6	$4\frac{3}{8}$	$5\frac{5}{8}$	3000	95	......	......	840	......	......	3	......	14.0
A. E. C. Acro	6	4.32	5.58	1500	77	83	500	850	1500	C	3	0.49	17.5
Blackstone	6	$4\frac{5}{8}$	6	1000	60	79	420	600	1250	E	1 (*b*)	0.42	28.5
English Electric	8	6	8	1500	200	79	450	850	3000	D	1 (*b*)	0.39	18.0
Lister	4	$4\frac{1}{2}$	$5\frac{1}{2}$	1000	54	82	450	750	2000	C	1 (*b*)	0.43	53.0
Mirrlees, Bickerton and Day (Ricardo)	3	$5\frac{5}{8}$	$6\frac{1}{2}$	1200	60	82	480	830	......	A	1 (*a*)	0.40	45
Continental Engines:													
Benz	6	4.13	6.5	1300	70	92	......	......	860	C	2	0.46	20.5
Deutz	6	4.53	6.69	1250	80	80	560	......	1080	C	2	0.46	19.8
Junkers‡	6	4.72	15.75	1600	600	90	......	......	......	A	1 (*a*)	0.37	3.1
M.A.N.	6	4.72	7.09	1400	90	81	......	600	4200	A	1 (*a*)	0.43	35.0
Saurer	6	4.32	5.9	1600	80	76	455	570	1150	C	3	0.49	12.5
American Engines:													
Buda-M.A.N.	6	$6\frac{1}{2}$	$8\frac{3}{4}$	1000	180	81.8	......	......	......	B	1 (*a*)	0.435	30.3
Cummins	6	$4\frac{1}{2}$	6	1000	60	83	500	600	......	S	1 (*b*) *S*	0.47	36.7
Hill	6	5	7	1000	75	72	......	......	1500	C	2	0.42	50
Packard	9	$4\frac{13}{16}$	6	1950	225	93	550	1200	6000	A	1 (*a*)	0.46	2.27

* A. Open, single hole.
B. Open, multi-hole.
C. Spring loaded, single hole.
D. Spring loaded, multi-hole.
E. Mechanically operated, multi-hole.
S. Special operated, multi-hole.

† 1 (*a*). Direct injection, port swirl.
1 (*b*). Direct injection, piston swirl.
2. Ante-chamber.
3. Auxiliary air chamber.
S. Special.

‡ 2 stroke-cycle, opposed pistons.

CHAPTER XXXVI

BURNING BY-PRODUCT FUELS AND REFUSE

453. By-Product Fuels.—(a) The **combustible materials** that remain after the main products of an industry have been produced are called by-product fuels. In most instances these by-products are burned by the plant in which they originate; but in a few cases they have a more extended use.

The sources, heating values, and average percentages of moisture of the most common by-product fuels in this country are given in Table LVIII. Note that the heating values are given on the basis of

TABLE LVIII

By-Product Fuels

Plant	By-Product Fuel	Gross Heating Value of Dry Fuel		Moisture in Fuel as Fired, Per Cent by Weight
		B.t.u. per lb.	B.t.u. per cu. ft. (68 deg. fahr. and 29.92 in. Hg)	
Blast furnace	Blast furnace gas	....	85 . . . 110	5 . . . 8 †
Saw mill Planing and finishing mills Automobile body and furniture factories	Wood refuse Hogged wood Sawdust	7,500 . . . 9,500	....	5* . . . 70
Coal gas generator	Coke and coke breeze	12,000 . . . 13,000	....	0.5 . . . 11
Cane sugar mill	Bagasse	8,000 . . . 9,000	....	40 . . . 55
Tannery	Spent tanbark	9,500	....	65
	Chestnut-extract chips	8,500	....	65
Petroleum refinery	Coke	14,000 . . . 15,000		
	Acid sludge	6,000 . . . 13,000		
Pulp mill	Sulphite liquor			

* Kiln-dried lumber.

† This is in the vapor form, the amount of which depends upon the humidity of the air used in the blast furnace and the method of washing the gas.

dry fuel, from which the heating value " as fired " may be easily found. For example, from the values given in the table for spent tanbark the heating value of this material, as fired, would be 9500 (1 − 0.65) = 3325 B.t.u. per lb.

(b) **Blast furnaces** normally produce about 6 tons (or roughly 140,000 cu. ft.) of **by-product gas** for each ton of pig iron tapped. Since the average furnace produces 600 to 800 tons of pig iron per 24 hours, an enormous quantity of gas is available for use. The combustion of the coke in the blast furnace is supported by means of highly preheated air, and the resulting gas contains a large percentage of nitrogen. Furthermore, the gas has a high carbon dioxide content, resulting from the chemical reactions of the smelting process. An analysis of blast furnace gas therefore shows considerable non-combustible material.[1] Since the chief combustible ingredient is carbon monoxide, the heating value of the gas is low, as shown in Table LVIII.

In the average blast furnace plant about 30 per cent of the gas produced in the furnace is required for preheating the blast air, and the remaining 70 per cent is available for the production of power. The energy derived from about one-fifth of this remaining gas (or 14 per cent of all the gas produced) is required to operate the blowers for the blast air. Thus approximately 56 per cent of all the gas produced by a blast furnace is available for electric power generation. An energy balance given by Streeter and Lichty [2] shows that for a blast furnace having a capacity of 500 tons of pig iron every 24 hours, the gas available per hour, for all power purposes, contains about 193 million B.t.u., measured above the standard datum. When the blast furnace plant is operated in connection with a steel mill, the electric energy produced from the by-product gas is used throughout the mill, and any surplus is usually fed into the nearest public utility network.

That portion of the by-product gas which is available for general power purposes may be used in steam-generating units [3, 4] or in gas engines.[2] However, there is an increasing tendency among operators to divert part of this gas and enrich it with other gases for use in heating industrial furnaces.[5] In any case, the gas is first cleaned, since it contains a large amount of dust when it leaves the blast furnace. If the gas is to be burned under boilers its dust content should not exceed

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[1] See Sect. **368** (**b**).

[2] Streeter and Lichty, "Internal-Combustion Engines," McGraw-Hill Book Co., 3rd Ed., 1929.

[3] Clarke, C. W. E., "Power Production in the Steel Industry," Proc. Second World Power Conf. (1930). See also Power, v. 72 (1930), pp. 263–264.

[4] See Blast Furnace and Steel Plant, v. 18 (1930), p. 986.

[5] See Blast Furnace and Steel Plant, v. 17 (1929), p. 857, and v. 18 (1930), p. 1476.

0.5 grain per cu. ft.; but if it is to be used in engines, an additional cleaning is necessary to reduce the dust content to about 0.005 grain per cu. ft.

When the gas is burned under boilers it may be used alone or in combination with oil or powdered coal. Since the gas has a low heating value and burns slowly, the combustion space within the steam-generating unit will be large, for a given capacity.

The gas engines used in blast furnace plants are large slow-speed horizontal machines with double-acting cylinders arranged in tandem.[2, 6] Although the overall thermal efficiency of a gas engine plant of this kind is quite favorable, the physical characteristics of these engines limit their use to large steel mills which operate a considerable number of blast furnaces. Polson [6] has reported test data for several of these large engines.

(c) **Wood** is one of the most important by-product fuels, and many modern finishing mills consume all of the mill refuse in the generation of steam for power and for kiln-drying purposes. The design of the furnace for burning wood depends upon the moisture content, size, heating value, and packing tendency of the wood and also upon the draft conditions and desired capacity. Dry sawdust, dry shavings, and dry " hogged " wood (i.e., wood refuse that has been passed through a shredding machine producing small particles of fairly uniform size) are burned in suspension in furnaces resembling pulverized-coal installations. The wood particles are usually blown into a bin, and fed to the furnace by means of heavy screw conveyors, the speed of the screws being controlled by the steam pressure.

In burning wet wood, the furnace is commonly arranged with a primary combustion space, or ignition arch, for drying and ignition, and a second combustion chamber in which the combustion of the gases is completed. If a large percentage of the wood fuel consists of edgings, blocks, and sticks, burning in suspension is impossible, and grates on which the larger pieces burn must be added to the furnace.

If the supply of wood refuse is not sufficient to carry the required steady load, an auxiliary fuel must be used. Frequently, coal and wood are burned in the same furnace; the coal is fed by mechanical stokers, and wood pieces are dumped into the furnace through the roof and burn on top of the coal. If sawdust, shavings, or " hogged " wood are used in conjunction with coal, they may be spouted through the front wall and burned in suspension over the coal.

The *temperature* within a wood-burning furnace may become very high, especially if the wood is fairly dry and only a small amount of

[6] Polson, "Internal Combustion Engines," John Wiley & Sons, 1931.

excess air is used with refractory walls. If no heat were absorbed from the furnace the products of combustion of wood would attain temperatures, as calculated by Gladden,[7] equal to those given by the curves in Fig. 655. These values serve to emphasize the necessity of cooling the refractory in a wood-burning furnace; both air-cooled and water-cooled walls have been successfully used. The preceding reference gives considerable detailed information regarding the design of furnaces for various kinds of refuse woods.

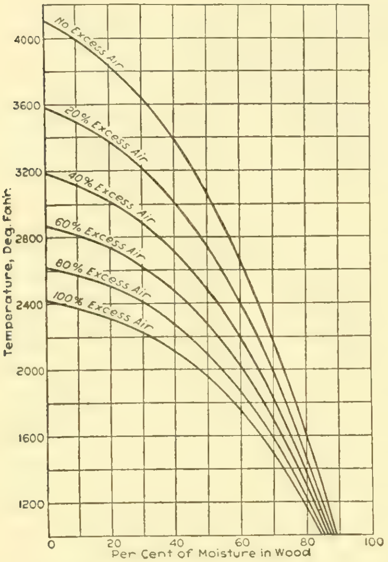

FIG. 655.—Furnace Temperatures Attained by the Adiabatic Combustion of Wood.

(d) **Coke breeze** is a by-product of coke manufacturing (Sect. **353**) and is difficult to burn. The forced-draft traveling grate (Sect. **414**) is about the only type of stoker suitable for burning this fuel.

(e) **Bagasse**, which is the residue left from the sugar cane after the juice has been extracted, may be burned in a furnace of large volume. The best results are obtained by using a primary furnace that is lined with refractories to maintain the temperature when the green fuel is introduced through the roof. In this primary furnace the fuel is dried and partially burned; the products from this chamber pass to the secondary furnace where additional air is admitted and the combustion completed before the products reach the boiler heating surface. High combustion rates are desirable when burning such a wet fuel.[8]

(f) **Spent tanbark** is the fibrous part of the tanbark left after use in the tannery, and carries a very large amount of water. Consequently it is difficult to burn, but combustion can be secured in a furnace similar to the one used for bagasse.

(g) **Petroleum coke, acid sludge, tank bottoms, cracking plant residue,** and **gas** are the principal by-product fuels from an oil refinery. In some refineries all five of these fuels are burned simultaneously.[9] The ash content of the petroleum coke is nearly negligible, but the moisture is often high. The furnace for burning refinery residues should always have its side walls provided with a sufficient number of access doors of the explosion type to relieve the pressure in case of explosions that are always likely to occur with such fuels. Some plants use natural gas or fuel oil as the main fuel and refinery waste as an auxiliary one. Recently one station [10] has been successful in using mechanical atomization of the refinery waste sludge combined with fuel oil. The mechanical atomization not only improves the efficiency of the steam-generating unit over that attainable from steam atomization, but the resulting flue gases have a lower dew point and thus the risk of corrosion from the acid in these gases is considerably reduced.

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[7] See "Boiler Settings for Burning Refuse Wood," Trans. A.S.M.E., FSP–53–3, 1931, p. 13.

[8] For additional information regarding special forms of furnaces for burning bagasse or wood refuse, see "Design of Furnaces and Fuel Feeders for Burning Refuse," by M. A. Hofft, Trans. A.S.M.E. Wood Ind.—53–1, Jan.–April, 1931, p. 1.

(**h**) **Sulphite waste liquor** is a by-product of paper making, and is produced in enormous quantities. Satisfactory methods of utilizing this by-product are eagerly sought. Some paper mills are burning the liquor in burners similar to those used to atomize oil.[11]

454. Burning City Refuse.—(**a**) The **waste** or **refuse** from cities consists of "ashes," rubbish, garbage, and sewage, all of which contain combustibles and may be burned with the proper equipment. The burning of sewage is difficult, but the gas evolved in the digestion tanks of sewage disposal plants is now used for the production of both heat and power.

Garbage usually contains so much moisture that it cannot be ignited or burned without being dried or mixed with rubbish, which is largely wood and paper. Such a mixture is called *mixed refuse* and usually contains 2 parts, by weight, of garbage to 1 part of rubbish.

When city waste is burned the heat is used to make steam and to dry the garbage. The amount of steam that can be produced per pound of refuse varies considerably with the type of plant and the material burned, but is usually from 0.5 to 1.5 lb. The steam is generally used to operate turbine-generators, and the electric energy thus produced often yields a revenue that is more than sufficient to pay for all the expense of the incinerator. Another source of income resulting from the combustion is found in the cinders, which are used to make brick and concrete.

(**b**) The **gross heating values** of various waste materials from cities are generally about as follows:

Rubbish	4000 to 8,000 B.t.u. per lb.
Mixed refuse	2000 to 5,000 B.t.u. per lb.
Garbage, as received	1000 to 2,000 B.t.u. per lb.
Garbage, dry	8000 to 10,000 B.t.u. per lb.

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[9] See Power, June 2, 1931, p. 868.

[10] See Power, Jan. 13, 1931, p. 56.

[11] See Power, Jan. 21, 1930, p. 88.

The rubbish is relatively dry, but the garbage usually contains from 40 to 80 per cent moisture, and consequently before drying its heating value is very low.

(c) The **furnace** of one important type of **incinerator** consists of groups of fireboxes, or cells, placed on two opposite sides of the boiler. These cells are made of cast iron with double walls and bottom. The air used for combustion is forced between these walls and through perforations in the bottom into the furnace, under a pressure of about 10 in. of water; the air is thus preheated to about 250 deg. fahr. and the walls are kept from overheating. Each cell is fed with a charge of from 500 to 1000 lb. of material which is allowed to fall by the opening of a horizontal register on the top of the cell. The cell is large enough to burn from 2 to 3 tons of material before the ashes and cinders have to be removed.[12]

Before the refuse is delivered to the furnace it is dried by some means. One type of drier consists of large rotating cylinders in which a considerable portion of the moisture from the refuse is evaporated by coming in contact with the hot flue gases leaving the steam-generator.

(d) **Examples** of large, successful, and well-managed **incinerator plants** are afforded by those in the city of Paris. During the year 1928 one of these plants gave the following results: [12]

Refuse burned	190,000 tons
Amount of electric energy generated	28,500,000 kw-hr.
Amount of energy sold	25,200,000 kw-hr.

From these figures one may note that the amount of energy sold was equal to 88.5 per cent of that generated, or in other words the operation of this plant consumed only 11.5 per cent of the electrical energy generated by the burning of city refuse, no fuel at all being purchased. Many other examples of incinerators successfully used by cities of various sizes in Europe and America may be found in such literature as that published by the American Society of Municipal Engineers.

(e) **Sewage gas** is produced during the conversion of the hydrated solids of the sewage into an odorless sludge in a sewage-disposal plant. This is accomplished in digestion tanks by means of bacterial action. During the process, which for any one batch of solids takes about two months, a large quantity of gas is evolved, and this gas is now being used, both in this country and abroad, for heating purposes and the operation of gas engines at sewage-disposal plants.

The disposal system used in the first American installation, at Charlotte, N. C.,[13] produces approximately 1 cu. ft. of gas for each 200 gal.

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[12] See Engineering News-Record, Dec. 19, 1929, p. 974.

[13] See Power, May 7, 1929, pp. 736–738.

of sewage handled. This gas is composed of about 70 per cent methane, and calorimeter tests have shown its gross heating value to be very high (about 850 B.t.u. per cu. ft.). The composition and heating value are subject to considerable seasonal variation.

Operating figures for the installation mentioned above show that during a single month the high-speed engines, using sewage gas, produced one quarter of the total energy required to run the disposal plant, which serves a population of 35,000.

The fixed charges on the equipment for utilizing the gas are $222 per month, but this amount is more than offset by a saving of $500 per month in the cost of purchased energy obtainable at 1.5 cents per kw-hr.

ADDITIONAL REFERENCES

"Design and Operation of Hog Fuel Burning Equipment," by H. A. Bastian, Combustion, Oct. 1934, p. 25.
"A Modern Steam-Generator with Combination Pulverized Coal and Wood Firing by K. Heinrich, Combustion, Jan. 1934, p. 9.
"Making Garbage Light City Streets," by L. J. Schenk, Combustion, Feb. 1934, p. 7

ADDITIONAL REFERENCES ON STOKERS

"Mechanical Stokers," by J. G. Worker and T. A. Peebles. McGraw-Hill Book Co.
"Sources of Coal and Types of Stokers and Burners Used by Electric Public Utility Power Plants," by W. H. Young. Brookings Inst. Pamphlet Series, Vol. II, No. 2, 1930.
"Stoker Developments at the Detroit Edison Company's Delray Boiler House No. 3," by P. W. Thompson and F. J. Chatel, Trans. A.S.M.E. 1933, FSP–55–11, p. 171.
"Principles of Underfeed Combustion and Effect of Preheated Air on Overfeed and Underfeed Fuel Beds," by P. Nichols and M. G. Eilers, Combustion, June-July, 1933, p. 15.
"Underfeed Combustion, Effect of Preheat, and Distribution of Ash in Fuel Beds," by P. Nichols, U. S. Bu. Mines, Bull. 378.
"Lignite Burned on Traveling Grates at Univ. of Texas," by C. J. Eckhardt, Jr., and W. H. Wood, Combustion, July 1934, p. 8.
"The Economics of Preheated Air for Stokers," by R. E. Dillon and M. D. Engle, Trans. A.S.M.E., 1934, FSP–56–17, p. 879.
"Domestic Stoker Performance" (Tests) by T. G. Estep and D. C. Sayler, Mech. Eng. Sept. 1934, p. 547, 635.
"Characteristics of Large Hell Gate Direct-Fired Boiler Units" by W. E. Caldwell, Trans. A.S.M.E., 1934, FSP–56–2, p.65.
"The Effect of Tempering Coal on Its Combustion," by R. A. Sherman, J. R. Blanchard, and D. M. Demorest, Combustion, Dec. 1934, p. 18.
"Automatic Zoned-Air Control for Conners Creek Stokers," by T. Wilson, Power, March 1935, p. 129.
"Ten Years of Stoker Development at Hudson Avenue," by J. M. Driscoll and W. H. Speer, Trans. A.S.M.E., 1935, FSP–57–3, p. 49.
"The Application of Central Station Practice to Domestic Heating," by M. K. Drewry, Mech. Eng., Feb. 1935, p. 103.

CHAPTER XXXVII

COMBUSTION CONTROL

455. Controlling Combustion.—(**a**) The **purposes** of controlling combustion in any steam-generating unit using any kind of fuel are (1) *to regulate the generation of steam to suit* the demand, (2) to *hold the pressure* in the steam main at the desired value, which is usually nearly constant, and (3) *to maintain the proper proportion of air and fuel* going to the furnace to give the best results from the unit, all things considered. Of these three items the third one is the most important in controlling the efficiency of the unit and the cost of maintenance of the furnace. However, the maintenance of the proper proportion of air and fuel at all rates of firing is difficult to accomplish because completeness of combustion, furnace temperature, and exit gas loss are all involved. Thus, for example, when burning pulverized coal, oil, or gas, only 10 to 20 per cent excess air may be desirable at low rates of operation, but at high rates the excess air may need to be nearly twice as great to maintain the furnace temperature sufficiently low to preserve the brickwork; whereas with stokers 50 per cent excess air may be desirable at low rates of firing in order to insure complete combustion, and only 10 to 20 per cent excess may be sufficient at high rates. Hence any combustion control system should be able to vary the proportion of air and fuel to suit the complex factors involved.

(**b**) The **furnace draft,** or the pressure inside the furnace relative to that of the atmosphere surrounding the furnace, is an important factor in any system of combustion control. With forced draft and induced draft fans in use, the furnace draft may be varied over a considerable range as desired. The general practice is to hold this draft *slightly negative* in order to keep a small current of air passing into the furnace from the room. This will prevent overheating the stoker fronts and doors; it will also prevent dangerous gases or flames from blowing into the boiler room. However, if the furnace pressure is kept *much less* than atmospheric the infiltration of air through the setting may easily be so excessive that the stack loss becomes too high for good economy.

When the furnace pressure is held substantially equal to that of the atmosphere, the furnace is often said to operate on a "*balanced draft.*"

There is considerable difference of opinion among high-grade operating engineers as to the amount of furnace draft that should be used with stoker-fired plants. The kind of coal, the thickness of fuel bed, and the frequency of inspection of the fire are all involved in this question, but the majority of operators prefer the balanced draft, rather than a high furnace draft, such as − 0.5 to − 0.8 in. of water, which is sometimes used.

(c) The **variation of steam pressure** in the outlet of a steam-generating unit is a sensitive indicator of the rate of flow of steam relative to the rate of heat absorption by the unit. If the feed water is delivered to the unit at a constant rate and temperature, a drop in steam pressure shows that the flow of steam has increased relative to the rate of heat absorption; and thus the obvious remedy is to increase the rate of firing, assuming that the unit has not already reached its maximum allowable rate of combustion.

If the fuel is burning in a mass, as when a stoker is used, the **quickest adjustment of the combustion rate** is accomplished by changing the rate of air supply, which is readily effected by varying the damper openings with natural draft, or by altering the speeds of mechanical draft fans, when they are used. The adjustment of the rate of fuel feed should of course also be made as quickly as possible. When powdered fuel, oil, or gas is burned the rates of fuel feed and air supply should be changed simultaneously. In addition to regulating the rates at which the *fuel* and *air* are supplied, a proper *differential of gas pressure* must also be maintained through the unit to cause the products of combustion to flow past the various heating surfaces to the stack at the proper speed, and to maintain a furnace pressure best suited to the operating conditions. These factors can be controlled by hand or by automatic apparatus, but in any case the variation in steam pressure is the chief sentinel always on duty to show instantly what needs to be done; and with automatic control this variation in pressure also supplies a small force that is sufficient to initiate the required movement of some suitable control apparatus.

(d) **Manual control of combustion** is based mainly on the knowledge of the operator and on his observation of the available instruments. An example of an effective combination of suitable instruments is the **Bailey Boiler Meter,** which is designed to visualize the operating conditions and to facilitate obtaining the best results at all times by maintaining the lines on its charts in the proper relative positions. This meter is calibrated in connection with actual tests of the unit to which it is attached and is so adjusted that for best results the air-flow line (blue) will coincide with the steam-flow line (red) at all rates of operation, and the flue temperature line (green) will have a definite position. The instrument may also record the rate of fuel feed (purple) or the temperature of superheated steam. The fuel feed should of course be made to vary directly with the steam-flow and air-flow. The record of the flue gas temperature is a check on the condition of the boiler baffles and on the efficiency of the heat transfer through the boiler tubes. The Boiler Meter may be supplemented by other gages and recorders giving information regarding feedwater temperature, flue gas temperatures at inlet and outlet to economizers, drafts at various places in the unit, fan speeds, and CO_2 content in the exit gases. These instruments are commonly grouped together and mounted on panels for convenience.

The proper methods of operating a hand-fired furnace have been given in Sects. **406** to **410**; Sects. **387** and **388** discuss the significance of the CO_2 and O_2 content of the exit gases. *Hand control* is still very common with small furnaces, but since 1920 *remote control* and *automatic control* of various kinds of fuel burners have been rapidly developed and successfully applied to nearly all large and medium-sized plants.

Remote control is the only form of manual control that is at all suitable for large units, because the auxiliaries of such a unit are likely to be very far from the main control board, say 100 ft. or more. In such cases the operator observes his instruments and decides what should be done; he then simply operates push buttons that control the motors for the fans, fuel feeding, and dampers. If the operating force is highly skilled and supplied with the proper instruments, the results obtained with the remote control system are very satisfactory. This system, however, requires a larger number of skilled operators than the automatic system, and the control does not respond to rapid fluctuations in load as quickly as the automatic. The first cost of the remote control system is a considerable fraction of that of the automatic, which is nearly a negligible portion of the total cost of a large steam-generating unit. In a small plant, however, the cost of either is too large a percentage of the total to make its installation feasible.

456. Automatic Control Systems.—(a) The **main advantage** of automatic control is its ability to *act immediately* and *simultaneously* throughout the plant when the demand for steam changes suddenly. This produces (1) a saving of fuel due to better combustion and smaller safety valve loss, (2) a reduction in the operating force, and (3) a higher degree of safety when the load on a high-pressure station goes off suddenly. When the automatic system is used, the attendants are relieved from the close regulation which is necessary without such apparatus, especially when the loads vary rapidly and frequently. Hence automatic control enables closer attention to be given to the condition of the fire and to the irregularities in operation arising from such things as changes in quality of fuel, inaccuracies in fuel feeding, clinkering, and slagging. Although highly efficient regulation can be obtained with hand control, the expert and close attention required is irksome and expensive; hence economy alone is usually sufficient to justify the use of automatic control apparatus, not only in large central stations but also in many medium-sized plants. In all cases, however, the apparatus should be under the supervision of *competent attendants.*

(**b**) The **primary actuating forces** that enable an automatic system to function are produced by (1) changes in steam pressure, (2) fluctuations in the rate of steam flow, (3) variations in the furnace draft, or (4) a combination of (1), (2), and (3). These primary forces are very small and are therefore only sufficient to operate some sensitive element in an apparatus generally called the *master controller* or *master regulator.* This master must be able to utilize these primary impulses to control stronger forces that are relayed mechanically, hydraulically, pneumatically, or electrically, to the individual controllers. These controllers regulate (*a*) the amount of opening in the air ducts to the furnace, or the speed of the forced-draft fans, thus controlling the amount of air to the furnace, (*b*) the outlet damper, or the speed of the induced-draft fans, thus controlling the draft within the unit and discharge of the products of combustion, and (*c*) the rate at which fuel is fed to the furnace.

(**c**) **Many different designs** of automatic systems are on the market, and each manufacturer will furnish the combination of his own apparatus that seems most suitable for the particular requirements of any plant. Where several steam-generating units are operating together, each one may be individually controlled, or there may be a master controller for the whole plant with such automatic or hand regulation at each unit as may be necessary to distribute the load as desired. In all installations of automatic control, provision must be made for changing from automatic to hand control in an emergency, such as the failure of supply of fuel, air, or feedwater. Furthermore, to insure reliability of automatic systems two or more independent sources of the medium that operates the various regulators are provided; and a quick shift from one source to another must be made when necessary.

(**d**) **A description** of all the automatic control systems in use cannot be included here, but a few of the representative ones used in this country will be given very briefly.

In the *Enco Balanced Draft System* (The Engineer Co.) each steam-generating unit is individually controlled. A change in the rate of the steam outflow alters the steam pressure which in turn controls the speed of the forced draft fan and thereby regulates the air supply and combustion rate. The resulting change in draft pressure in the furnace actuates a diaphragm which controls the damper or induced draft fan in such manner as to maintain a constant predetermined draft-pressure in the furnace, and also adjusts a cam which regulates the fuel-feed.

In the *Smoot Control System* (Smoot Engineering Corporation) a regulator is connected with each function to be controlled, such as on the fuel feed, the forced- and induced-draft fans, air gates, stack dampers, and also to the feed pump, if pressure-differential controlled; and all these regulators are connected to the master controller through a compressed-air system. The master is operated by changes in the steam pressure, and through a leak-valve alters the air pressure on diaphragms in the various regulators. Each regulator has a second diaphragm which compensates for the local variations in the factor controlled by it. The master controller contains mercury manometers which show the ratios existing between the various factors under control and guide the operators in making adjustments.

In the *Hagan System* a change in pressure in the steam main actuates a *master regulator* which controls simultaneously, either by compressed air or by mechanical means, (1) the dampers on all the units, (2) the fuel feed to all furnaces, and (3) the speed of the forced-draft fans. Each unit also has a combustion controller, which is actuated by draft variations in the furnace and which adjusts the wind box damper to compensate for varying fuel bed resistances and thus maintains the correct draft over the fires. When a central control is used, low-pressure air under a partly submerged bell, which is raised or lowered by the variations in steam pressure, operates in unison similar bells in the individual controls. Each control has a second bell which is affected by the factor regulated and compensates for differences between the local condition and the average at the master.

Among the *electrically operated controls*, the operation of the *Bailey Meter Control* may be considered equivalent to a man constantly watching a pressure gage and pushing buttons every five or ten seconds to energize remote controlled motors which make the proper adjustments at the various units. In the *master contactor* a small motor continuously operates a stepped cam which intermittently comes into contact with a presser pin that is shifted to different steps by a Bourdon tube when the steam pressure changes; thus the length of contact depends on the rate of change in steam pressure from the standard. The electrical impulses sent out energize solenoids on clutches in the *control drivers.* Any deviation from the proper ratio between the air flow and the fuel feed is compensated for by a solenoid which is energized intermittently by a contact closing on the steam-flow air-flow meter.

The *Leeds and Northrup system* is operated by a direct electric current to which the combustion rate is proportioned. The master regulator consists of a steam-actuated rheostat which is in series with the operating circuit.

Also several other satisfactory automatic control systems are made in this country. For a more detailed discussion of the various systems further references[1] are given.

457. Banked Fires.—(**a**) The **meaning** of the term, *banked fire* (or less correctly, *banked boiler*) is that the rate of combustion has been purposely reduced to a very low rate, or even entirely stopped, for some period of time during which the demand for steam has become very low or has entirely ceased. Thus in a small plant which may operate only 8 hours daily the banking period would be approximately two-thirds of the time. In central stations, reserve units must be held ready to deliver steam quickly in case of a sudden and large increase in the output of the station.

The terms **dead bank** and **floating bank** are often employed. As used by Thompson,[2] in connection with stoker-fired units, a *dead bank* means that " the coal feed is entirely shut off and the fire is allowed to burn back as far as possible without permitting it to go out entirely; a *floating bank* means that the coal feed is sufficient to keep the tuyeres well covered and to hold a fire which will maintain full pressure in the boiler." As only sufficient coal is fed to the furnace to keep the fire during a dead bank, the steam pressure may even fall to atmospheric if this bank is continued long enough. On the other hand, the boiler will probably deliver steam at a very low rate during most of the time that the floating bank is maintained.

(**b**) The **banking loss** means the amount of fuel used to maintain a floating bank, or to maintain a dead bank and then bring the steam pressure back to normal. Naturally the banking loss depends upon the duration of the banking period, the size of the unit, the kind of fuel used, the type of banking, the air leaks past the dampers and through the setting, and the insulation of the entire steam-generating unit. The loss during a dead bank is somewhat smaller with powdered coal, gas, or oil-fired units than with stoker-fired ones of the same size, because the former units may have their dampers tightly closed after the fuel feed is shut off and thus the loss due to the circulation of air through the furnace is prevented. To have a dead bank with a unit fired with solid coal, the dampers must not be completely closed, as the furnace would then be converted into a gas producer and an explosion hazard would be created. Such a unit has not only the radiation and convection losses from its entire outer surface, as in the other cases, but also the additional losses due to the hot gases and unburned combustible passing out the stack, and the unconsumed carbon going to the ash pit.

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[1] "Automatic Combustion Control," by C. H. Sanderson and E. B. Rickets, Trans. A.I.E.E., Dec. 1931, Vol. 50, p. 1378; "Combustion Control," by T. A. Pebbles, Mech. Eng., March, 1925, p. 193; "Combustion Control for Boilers," by R. J. S. Pigott, Mech. Eng., October, 1924, p. 590. Also the following in Power: May 6, 1922, p. 771 (Smoot); May 11, 1926, p. 711 (Enco); Aug. 10, 1926, p. 198 (Bailey); Oct. 26, 1926, p. 626 (L. and N.); May 31, 1927, p. 814 (General); Nov. 22, 1927, p. 770 (B. M. T.); Sept. 9, 1930, p. 412; Sept. 1, 1931, p. 314 (Powerton); April 19, 1932, p. 584 (Purdue); and May 31, 1932, p. 809 (Hudson Ave.).

[2] See "Tests of a Type W Stirling Boiler," by Paul W. Thompson, Trans. A.S.M.E., 1922, p. 1015; also "Coal Charges for Banking, Lighting and Burning Out," by G. C. Eaton, A.S.M.E., Dec. 1934.

In bringing a unit back into operation a certain amount of fuel must be consumed before all parts are returned to their normal operating temperatures, and this loss occurs with all methods of firing. Generally, when a unit is banked for a considerable period of time, as over night, and especially when the equipment may have to be brought into service again quickly, the powdered-coal firing has considerable advantage over stoker firing. For example,[3] the powdered-coal-fired steam-generating unit shown in Fig. 545 was shut down for 9 hours, with feeders and fans stopped and dampers closed, and then in 10 minutes was made to generate steam at the rate of 100,000 lb. per hour; and from a dead cold condition this unit was brought-in on the line in 30 minutes, as compared with two or three times as long with stoker-fired boilers in the same plant.

The amount of **fuel consumed during a banking period** (including that required to bring the unit back to normal) is difficult to determine in general, because so many factors, as previously given, are involved. Furthermore, many of the tests have not included sufficient data to enable comparative results to be obtained. However a few values are available from various tests[4, 5] on stoker-fired units and, when reduced to the common basis of unit area of grate surface, give the following results:

Length of Banking Period, in Hours	Coal Required, in Lb. per Hr. per Sq. Ft. of Grate
10	3.0 to 5.0
20	2.0 to 4.0
30	1.5 to 2.5
40	1.5 to 2.0

[3] See Power, Aug., 7, 1928, p. 225.

[4] See "Tests of a Type W Stirling Boiler," by Paul W. Thompson, Trans. A.S.M.E., 1922, p. 1015. [5] Proc. N.E.L.A., v. 84, 1927, p. 954.

ADDITIONAL REFERENCE

"Automatic Combustion Control for Boilers," by J. L. Hodgson and L. L. Robinson, Inst. of M. E. 1934, also in Engineering, Feb. 2, 1934, pp. 129, 134, 162; and in The Engineer, Feb. 2, 1934, pp. 125, 128, 132.

CHAPTER XXXVIII

FURNACES

458. General.—(a) The furnace of a steam-generating unit has for its *primary purposes* the provision of (1) places for the fuel burner or burners to function and for the combustion air to be supplied, (2) an enclosure to confine the gases and to provide suitable space and environment for completing the combustion process, (3) means for utilizing as fully as is expedient the energy made available within it, (4) an outlet for discharging the products of combustion, and (5) means for the voidance of the refuse. The *furnace space* is surrounded by the *furnace walls*, that, with their related parts, constitute the *furnace setting*, which is generally also a part of the *boiler setting*.

(b) The **design of the furnace** of a steam-generating unit has an influence not only on (1) the efficiency of performance of the entire unit, but also on (2) the initial cost of the unit as a whole and that of the building occupied by it, and on (3) the operating expenditures for fuel, labor, and maintenance, and for the energy used for driving some of the auxiliaries, such as the draft fans; hence, in modern power-plant practice, especially that connected with large industrial plants and central stations, the closest attention is given to providing furnaces that are the most suitable for the conditions to be encountered. It has been largely due to the more intensive operation of steam-generating units, as made possible by better furnace design, that the first cost of steam power plants has been kept from advancing in recent years during which the costs of construction materials and labor have increased greatly.

(c) The **main considerations** influencing the design of a furnace are: (1) the kind, condition, preparation, and characteristics of the usual or alternative fuels to be burned; (2) the properties of any ash or clinker that may be formed; (3) the normal, minimum, and maximum loads to be carried, and their relative durations; (4) the size and relation of the furnace with respect to the boiler and other parts of the unit; (5) the kind of stoker or other burner to be used, and, with powdered, liquid, or gaseous fuel, the method of firing (see Sects. **425**, **437**, **445**, and **446**); (6) the excess-air coefficient which it is desired to use; (7) the permissible carbon carry-over with the fly-ash and with the cinder deposited in the ash pit; (8) the initial costs; and (9) the operating costs.

(d) In general, with a given type of furnace construction, the **size and cost** of the steam-generating unit needed for a designated steaming capacity may be decreased by employing a higher rate of heat-evolution per cubic foot of furnace volume; and the efficiency of the unit may be raised by curtailing, within reasonable limits, the amount of excess air supplied. Thus, there is a tendency to use, for a given rate of steam generation, as small a furnace volume and as high a furnace temperature as can be employed with the chosen wall construction and yet not have too great an expense for maintenance and repairs. Similarly, an existing unit will, of course, have its greatest safe output, combined with highest efficiency under that loading, when its furnace is operated as intensively and at as high a temperature as the walls will withstand. The heat-release rates and temperatures allowable depend, of course, on the wall construction employed.

(e) The usual **kinds of furnace walls** are (1) solid masonry walls of refractory material—known as *solid refractory walls*; (2) *hollow air-cooled refractory walls*, (3) *bare water-cooled metallic walls*, and (4) *covered water-cooled metallic walls*.[1]

In this list the different kinds of walls appear in the order of (1) decreasing volumes of furnace required for a given output, (2) increasing rates of heat release allowable per unit of furnace volume, and (3) temperatures permitted, as far as the wall itself is concerned. The usual values of allowable energy-release rates are given in the next section. Ordinarily, the simple solid-wall construction is the most economical one for furnaces which are to operate always under moderate conditions; water walls are necessary when prolonged intensive rates of firing and high furnace temperatures are to prevail; and air-cooled refractory walls, or walls combining refractory and water-cooled sections, may be the most suitable for intermediate conditions.

(f) Although the **temperature** within a given furnace can, of course, be readily reduced by using more excess air, such practice lowers the efficiency of the steam-generating unit and should be avoided in normal operation; however, excessive dilution may be permissible under emergency conditions and while meeting peak loads which are of short duration. With other conditions the same, a higher furnace temperature results when preheated air is used for combustion than when air at room temperature is used. If a furnace is to operate intensively at high temperature over considerable periods of time, provision to withstand these conditions should be made in the original design.

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[1] Sometimes superheater or reheater surface takes the place of some of the wall surface that otherwise would be cooled by water. The terms "water-cooled walls" or "water-walls," when used in a general way, will be understood to include these other surfaces.

(g) The **limiting temperature** to which the reacting agents within the furnace may be raised without detriment to the furnace walls is chiefly dependent on the material of the original wall itself; but this temperature is also greatly influenced by the *fusing temperature and the composition of the ash* in the fuel burned. Refractory wall materials may be altered by slag penetration or by chemical reaction with the molten fly-ash deposited thereon, and may be eroded by liquid slag flowing over their surfaces. *If conditions could be kept uniform, the best refractory wall would be one that could be maintained, without deterioration, at a surface temperature just equal to the fusing temperature of the fly-ash.* But conditions vary, and if such a wall were subjected to higher temperatures under more intensive operation it would be gradually washed away. A somewhat colder wall may initially have a detrimental effect on the combustion process, but eventually solidified fly-ash may build up on it until the combined thickness becomes such that the temperature at the surface equals the ash fusing temperature; then, with the changes in the furnace temperature, the slag coating will increase or diminish in thickness in such manner as to maintain an equilibrium temperature at the surface. With refractory-faced walls, either air-cooled or water-cooled, the refractory itself can thin down, or the slag deposit build up, to reach the equilibrium condition. The least difficulty from wall slagging is encountered with metallic walls which are either water-cooled or steam-cooled, but no metal or refractory has been found that will successfully resist continuously the erosion of molten slag streams, even though the melting point of the material may be higher than the fusing temperature of the ash; hence, extensive slag flow in direct contact with the surface of the wall is to be avoided.

(h) In general, a certain **arrangement and construction of furnace** will give the best return on the initial investment and operating costs for each fuel, each combination of elements forming the steam-generating unit, and each set of operating conditions. So many variables are involved that standardized designs have but limited application, hence it is usually necessary to consider each installation as a special case. If coal is the fuel to be burned, it is desirable to have available an analysis of the particular kind (or kinds) of the coal to be used, and also to have information regarding the properties of the slag. Lacking this information, however, the discussion and data given on pages 324 to 330 will be of value. The special arrangements of furnaces for hand and stoker firing and for burning powdered, liquid, and gaseous fuels have been considered in general in previous sections.[2] It remains to discuss in this chapter the furnace volumes required (or heat-release rates permitted), the usual types of furnace construction, the available wall materials, the transmission of heat to and through the walls, and some details entering into furnace design.

459. Furnace Volumes; Rates of Energy Release.—(**a**) The **volume of a furnace** required in any given case depends upon the total amount of energy desired from the furnace per unit of time, and upon the **specific energy release-rate.** The latter term is commonly expressed in B.t.u. per hr. per cu. ft. of furnace, and depends on such factors as the flame length, ash fusion temperature, method of firing, excess air percentage desired, degree of turbulence to be provided, and type of furnace construction. With hand or stoker firing the size of the furnace cross-section, measured transverse to the direction of flame travel, is usually fixed by the area of the grate surface; and, when the fuel is burned in suspension, it is often dependent on the projected area under the boiler proper. The height (or length) of the furnace is then made sufficient to give the volume necessary to provide for the length of flame travel.

(**b**) As **hand-fired furnaces** cannot be operated very intensively, the distance from grate to boiler surface can be made relatively small, say 3 or 4 ft. with horizontal return tubular boilers (Fig. 537) and from 4 to 4½ ft. with horizontal water tube boilers (Fig. 554), when fired with volatile coal, and a little less with anthracite. In **stoker-fired furnaces** the distance that the evolved gases travel while burning is much greater than that with hand firing and, consequently, the boiler surface should be placed still farther from the fuel bed. The Pittsburgh Tests (Sect. **404**) give considerable information as to the flame length and the volume of combustion space needed with representative coals burned at various rates. The minimum heights at which stoker-fired boilers should be set, as recommended by the Stoker Manufacturers' Association, are given in Table XLIX, p. 486. But for very high rates of combustion per square foot of grate area, still greater heights may be desirable. Apparently, with mid-west coal, the maximum heat-release rate obtainable without smoke in stoker-fired furnaces of small firebox boilers and horizontal return-tubular boilers is about 50,000 B.t.u. per hr. per cu. ft. With refractory walls and underfeed stokers, the rate should not exceed 40,000, although double this value has been used, but probably with a relatively high percentage of excess air. Better combustion is obtained with still lower rates; also, furnaces with arches exposed to intense heat may require the use of somewhat more moderate furnace conditions. Even with water-cooled walls, a rate above 50,000 is not often allowed. The relatively high rates of energy release used in furnaces fired by hand or by stoker, as compared with furnaces in which the fuel is burned in suspension, is permissible mainly because the proportion of excess air supplied is generally larger than that used with the latter method of firing. With high rates of combustion per square foot of grate, the fly-ash problem with certain coals may require the use of low energy-release rates, but not as low as those to which furnaces for burning the coal in powdered form are limited.

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[2] For a general discussion of the combustion space needed in coal fired furnaces, see Sects. **402** (p. 464) and **404** (p. 465). For specific furnace requirements, see: for hand firing, p. 473; for chain grates, pp. 488–491; for underfeed stokers, pp. 502, 503; for powdered coal, pp. 510, 530–533; for liquid fuel, p. 566; for gases, p. 551.

(c) **Pulverized-coal furnaces** require, in general, larger volumes, or lower specific rates of heat release, than do furnaces of equal capacity using any other form of fuel. Conservative **maximum values of heat-release rates,** in B.t.u. per hr. per cu. ft. of furnace volume, for the different wall constructions and with fuels having various ash fusion temperatures, are given in Table LIX, which is based on recommendations of various manufacturers of equipment for burning powdered coal. Though not so stated, it is probable that the values given are mainly for vertical firing. The effect of the method of firing on the maximum safe combustion rate permissible with various types of walls is indicated by the values given, in the same units as before, in Table LX, due to DeLorenzi, and presumably are for coal having a fairly high ash fusion temperature. The rates to be used should, of course, be selected with respect to the particular combination of factors involved; and, although

TABLE LIX

ENERGY-RELEASE RATES *

Wall Construction	Ash Fusion, 2100 deg. fahr. and Below	Ash Fusion, 2100–2400 deg. fahr.	Ash Fusion, 2400 deg. fahr. and Above
Solid walls	Never	12,500	15,000
Air-cooled, without water-cooled bottoms	12,000	15,000	17,500
Air-cooled, with water-cooled bottoms	16,000	17,500	20,000
Water walls	17,500	22,000	30,000

* From Report of International Railway Fuel Association, Power, June 19, 1928, p. 1119.

TABLE LX

Energy-Release Rates *

Wall Construction	Vertical Downward Firing	Horizontal Turbulent Firing	Horizontal Tangential Turbulent Firing
Solid refractory walls and no cooling in furnace bottom	14,000	16,000	
Solid refractory walls with water screen or water-cooled ash hopper	18,000	22,000	
Air-cooled walls	21,000	23,000	
Fully water-cooled walls	25,000	28,000	34,000

* From Power House, Sept. 20, 1929.

values exceeding those tabulated have been used, attempts to go higher may lead to difficulties which are expensive to remedy.

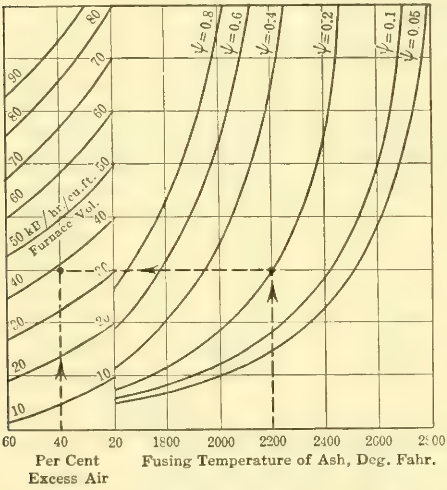

Fig. 656.—Approximate Relations Between Heat Release Rates, Excess Air Percentages, Fusing Temperatures of Ash, and "Fractions-Cold" (ψ). (kB = 1000 B.t.u.)

In general, the smaller the steam-generating unit, the greater will be the permissible specific release rate in the furnace, other conditions being the same, as the ratio of wall surface to furnace volume is greater in small furnaces than in large ones.

(**d**) The relation between specific release rates, excess-air percentages, fusing temperatures of the ash, and different fractions (ψ) of the furnace walls cooled by boiler tubes and water walls, is indicated in a very general way in Fig. 656, due to E. G. Bailey.[3] Even though the chart is undoubtedly inaccurate, it is useful for showing (although very approximately) the influence of the various factors involved. The quantity ψ is commonly termed the "fraction cold," and is discussed in Sect. **467.**

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[3] "Some Factors in Furnace Design for High Capacity," Tr. A.S.M.E., 1928, FSP-50-78, p. 253. The curves are based upon the assumption that all parts of the wall surface are maintained at the same temperature, the value of which equals the ash fusion temperature. Actual walls, however, do not function under such ideal conditions, hence the release rates may be lower than the values given by the chart, the solid and air-cooled refractory walls departing therefrom more than do water-cooled refractory walls.

(e) For **oil-fired and gas-fired furnaces,** the size and proportions of the combustion spaces should be based largely on the length, path, and spread of the flame. Furnaces for oil-firing are discussed in Sect. **446,** p. 566, and those for natural gas-firing, in Sect. **437,** p. 551.

The furnace for burning **blast furnace gas** should be of greater volume than that needed for natural gas. This is because this gas, which is of relatively low heat value, is slow to ignite and, owing to the presence of a large percentage of inert constituents, is more difficult to mix properly with the necessary air. Turbulence—provided by supplying the air under pressure, by projecting a multiplicity of streams of gas and air so that they intermingle, or by tangential firing—makes permissible using greater release rates and obtaining higher efficiencies than are obtainable with non-turbulent firing. But the dust content of this gas also has an important bearing on the design of the furnace and likewise on that of the burners and boiler. The chemical composition of the dust is such that it may remain plastic and gummy at temperatures as low as 1400 deg. fahr., and the slag formed adheres tenaciously to refractory surfaces. With dust-laden gas, it is necessary to prevent, wherever possible, the flames from making contact with high-temperature refractory surfaces, or sweeping over accumulations of dust on the furnace bottom. Cleaning the gas is of course beneficial, but is an added expense. According to DeLorenzi,[4] the heat release with *refractory walls* should not exceed 12,000 B.t.u. per hr. per cu. ft. when the gas contains more than 0.5 grain of dust per cu. ft.; but with less dust content than this, the rate may be as high as 18,000. *Water-cooled furnaces* with water-screened bottoms and tangential firing have operated satisfactorily with release rates almost double these values. The water walls are self-cleaning, for the deposit of sintered dust continually breaks loose and falls into the screened refuse hopper from which it is readily removed.

(f) In **conclusion,** it should be noted that the heat-release rates considered in the foregoing discussion are *average* values for the whole effective furnace volume. Actually the energy distribution throughout the space is not uniform and some parts of the wall surfaces may be at temperatures 300 or 400 deg. above the mean; thus, when the mean is allowed to be near the limit of safe operation, this added temperature may cause localized difficulty. In a thorough analysis, therefore, the conditions to be encountered by each element of volume and by each unit of wall surface should be investigated.

[4] "Corner Firing of Blast Furnace Gas," Combustion, Jan., 1932, p. 11.

460. Furnace Walls of Solid Brickwork.—(a) The walls of the more moderately operated furnaces of externally fired boilers are commonly of solid brickwork, and generally constitute a portion of the wall setting of the boiler itself. A great variety of brick wall constructions are used, of which only a few of the more typical ones can be shown here. Furnace settings of this kind are lined with good-grade firebrick or refractory blocks, the outer portions of the wall being usually of common red brick, second-grade firebrick, insulation, or a combination of these; sometimes the walls are enclosed in a steel casing.

(b) Simple **furnace walls suitable for Horizontal Return Tubular Boilers**, such as that illustrated in Fig. 537, are shown in Fig. 657.

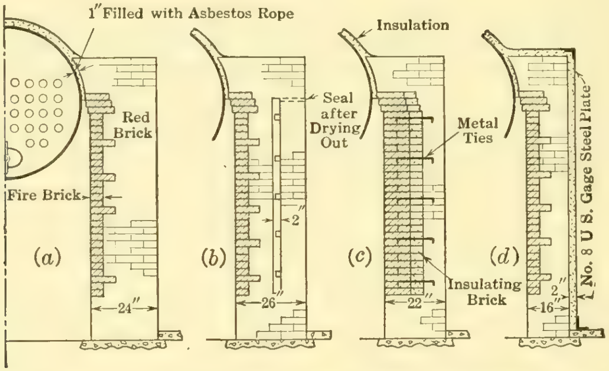

Fig. 657.—Some Forms of Furnace Walls for H.R.T. Boilers (as recommended by the Hartford Steam Boiler Inspection and Insurance Co.).

The walls are shown in the order of their relative costs. Although the solid wall (*a*) is least expensive to construct, it tends to crack through and thus to allow air infiltration which decreases the efficiency of the unit. With the double-wall construction, (*b*), the outer wall remains intact, and the space between walls can be filled with ashes, sand, or other suitable loose material which will seal any cracks that may form in the inner wall. At (*c*) is shown a wall containing insulating brick which, while reducing the loss of heat from the exterior surface of the wall, causes the brick lining to be hotter; hence this lining should be made thicker and of better material than would otherwise be necessary. The wall shown at (*d*) has a steel casing, which is separated from the brickwork by a layer of 85 per cent magnesia or other insulating material. In such walls it may be desirable to use second-grade firebrick between the inner lining and the insulation.

The wall sections shown in Fig. 657 are taken at the rear of the bridge wall and are suitable for combustion rates up to 25 lb. per hr. per sq. ft. of grate surface. In the furnace itself the walls are usually made 6 in. thicker at the grate level and are battered, as in Fig. 658, which shows an arrangement of complete setting for a H.R.T. boiler. Note that, in the latter figure, allowance for expansion is provided at the ends of the bridge wall and that the blowoff pipe is protected from flame impingement. The *relieving arch* in the side wall permits the replacement of lining underneath without disturbing the wall above, but this arch is often omitted because it adds to the initial cost of construction. The walls should always rest upon rigid foundations, otherwise they may settle, become distorted, and crack, thus allowing detrimental air infiltration.

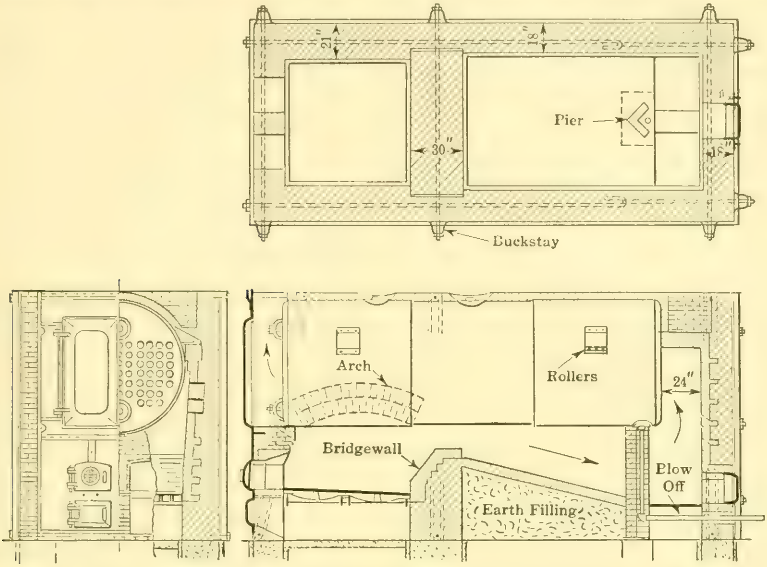

FIG. 658.—One Arrangement of Setting for a H.R.T. Boiler with Full Flush Front. Also see Figs. 537 and 715.

Jointless monolithic wall linings may be made by using plastic fireclay which is pounded or rammed into place. These linings should, of course, be securely anchored to the outer walls, three of the many methods used being shown in Fig. 659. Also, to allow for free expansion when heated, space should be left at the abutting ends of the lining, and, if necessary, this space can be filled with asbestos. At (*c*) in Fig. 659 is shown a section of an insulated and steel-incased setting in which

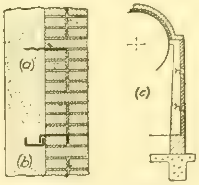

FIG. 659.—Monolithic Lining and Wall.

the wall proper is entirely of monolithic construction. Incidentally, plastic fireclay is also useful for patching or relining old walls and for making special shapes for new ones. Also, old and new walls may be protected on their inner surfaces, and their joints sealed, by coating them with a layer of highly refractory plastic material which may be troweled or brushed on, but is preferably applied with impact, as by means of a spray gun operated by compressed air or steam.

When fired by hand or by some kinds of stokers the furnaces of boilers of the H.R.T. type do not need to have access openings in their side walls, and in such cases the boilers in a plant can be arranged side by side in a continuous "battery," with walls in common between adjacent units. This arrangement reduces the number of walls needed and the space occupied by the installation. The furnace walls that are heated on both sides must be somewhat thicker than the exterior ones need to be.

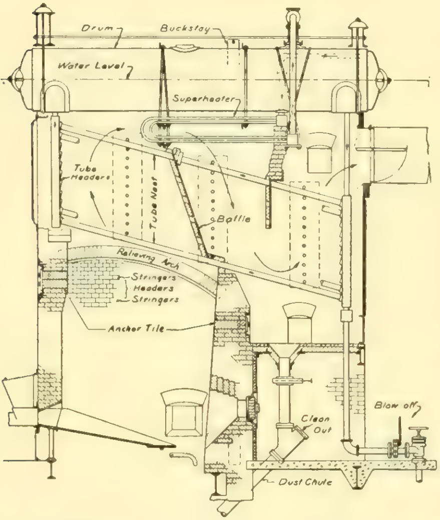

Fig. 660.—Example of a Furnace and Boiler Setting for a Unit Fired by an Underfeed Stoker.

(c) Solid brick walls for **stoker-fired furnaces** used with water-tube boilers are arranged in a great variety of ways, each depending mainly on the type of stoker and boiler used. Only two examples will be given in some detail here, and they illustrate only a few of the many possible constructions employed. Fig. 660 shows, in a general way, the arrangement of a unit of moderate size which is fired by means of an underfeed stoker, and which has a water-tube boiler with longitudinal steam drums and with tubes extending (nearly) horizontally between sectional headers that are connected to the drums by nipples.[5] This type is commonly referred to as having "horizontal" tubes. The boiler is suspended from overhead beams in such manner as not to be in contact with the walls and other parts even when distortions and expansions due to heating occur. The front wall and more highly heated portions of the bridge wall are wholly of high-grade firebrick. The side walls of the furnace and the rest of the walls of the boiler are lined with firebrick of suitable thickness and quality, the outer portion being of red brick. A poorer and cheaper grade of brick is used where permissible. The relieving arches in the side walls prevent the highly heated brickwork from carrying the weight of the parts above, and provide for the replacement of this lower portion without disturbing the rest of the wall. These arches must have rigid buttresses for taking the thrusts, and expansion space must be provided underneath them. To avoid the possibility of the front wall falling into the furnace after repeated heating, the upper part is anchored, by means of bonding tile, to the outside metallic parts of the furnace front. Departing from the simpler arrangement of bridge wall and space back of it, shown in Fig. 555 (p. 474), this setting has, at the rear, a raised floor (with dust-collecting pocket and chute) placed at an elevation that permits access to the inspection doors provided in the bridge well. As observation, access, and soot-blowing openings must be placed in at least one side wall of such units, these units cannot be arranged in continuous batteries of more than two. In the dual arrangement, the common wall between furnaces is made of solid firebrick of high grade and must be much thicker than the side walls. Several sections of the walls of a setting for a Babcock and Wilcox boiler of this type are shown in Fig. 661, which is mainly self-explanatory. Note particularly, however, the provision made for expansion and contraction of the various parts without interference; and that the steel work supporting the boiler should not be imbedded in the brickwork, as it might become overheated. In other solid-wall constructions, use is made, singly or in combination, of plastic refractory, insulating brick, or steel casings lined with insulation. It is the custom for the manufacturers who supply the boilers for a plant to provide also the necessary setting plans, the design of which is based on their previous experience with similar installations.[6]

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[5] For additional details of the stoker and furnace, see Fig. 585, p. 499, and for those of the boiler, see Fig. 725.

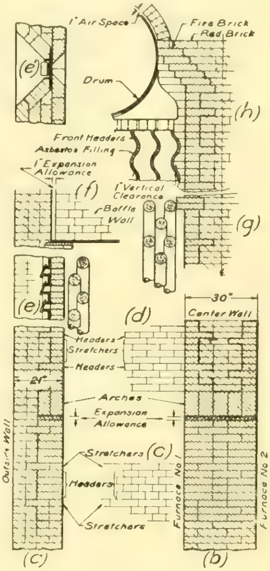

Fig. 661.—Various Sections of Furnace and Boiler Walls of Unit in Fig. 660.

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[6] For further discussion of brick settings, see the excellent chapter on this subject in "Steam," published by the Babcock and Wilcox Co.

(d) **Another solid-wall furnace setting** of interesting construction is shown in Fig. 662. This furnace, which is quite large and high, is designed for rapid rates of operation, and has a stoker so long that there is no bridge wall.[7] The walls are of refractory brick throughout, with special grades used in places where the punishment is greatest. The side and rear walls are concave so as not to bulge inward when heated, and are tilted outwardly 5 deg. from the vertical for the same reason and also to make less direct the heat impingement and the scrubbing action of the gases. The walls are backed-up by steel work on the outside, and each has a relieving arch, and is divided into panels, all with expansion joints between them. The parts of the walls (not shown) above the lower tube line of the boiler are supported independently of the parts below. Insulation is omitted from the furnace walls in order to prevent them from becoming overheated. Another illustration of relieving arches is given in Fig. 663,

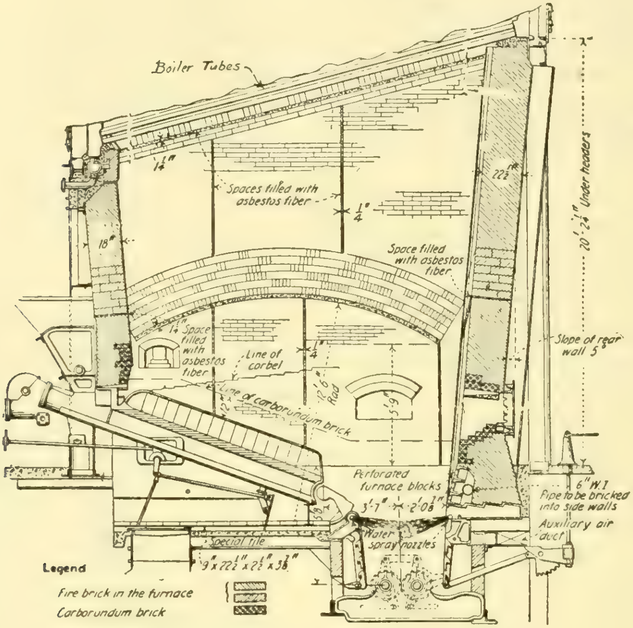
Boiler Tubes
$1\frac{1}{4}''$
Spaces filled with asbestos fiber
$\frac{1}{4}''$
18″
Space filled with asbestos fiber
$22\frac{1}{2}''$
$20'\text{-}2\frac{1}{4}''$ Under headers
$1\frac{1}{4}''$ Space filled with asbestos fiber
Line of corbel
$\frac{1}{4}''$
12′ 6″ Rad
5′ 9″
Line of carborundum brick
Slope of rear wall 5°
Perforated furnace blocks
$3'\text{-}7''$
$2'\text{-}0\frac{1}{8}''$
6″ W.I Pipe to be bricked into side walls
Auxiliary air duct
Water spray nozzles
Special tile
$9'' \times 22\frac{1}{2}'' \times 2\frac{1}{2}'' \times 3\frac{7}{8}''$
Legend
Fire brick in the furnace
Carborundum brick

FIG. 662.—Furnace Walls for 16,800 sq. ft. Boiler. (Devon Station, Connecticut Light and Power Co.)

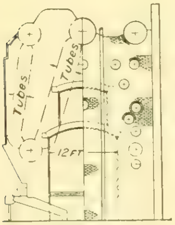

FIG. 663.—Relieving Arches in a Tall Furnace Wall. (Conner's Creek Plant, The Detroit Edison Co.)

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[7] For more details, see Power, May 13, 1924, p. 763, and March 25, 1924, p. 474; also Proc. N.E.L.A. 1924, p. 1324.

which shows the use of one arch above another. Under each of these arches is a concave panel of blocks which can readily be replaced.

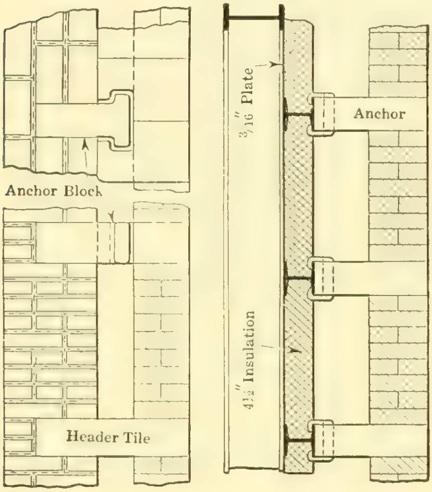
(e) The portions of **furnace walls adjacent to the fuel bed** in most instances need special consideration, as they are not only subjected to the heat of the fuel bed but also to the formation of projecting clinker adhesions which reduce the effective grate area and, with the moving beds present in stoker firing, scrape the sides of the bed and thus allow an excessive inflow of air at these places; further, these walls may be damaged when the adhering clinker is removed, as by using a slice bar. Thus, in such parts of the wall, it is often necessary to use special slag-resisting wall blocks (which may be either solid, as in Fig. 661, or hollow, as in Fig. 588), or to substitute air- or water-cooled metallic walls sections, as in Figs. 585 and 589.

(f) The **joints in refractory walls** are the places where the disintegration usually begins; hence they should be minimized and constructed with the greatest care. Joints should be made as thin as possible, using in them fireclay or special cements which should be at least of as high grade as the brick. The clay should be considered as serving merely as a filler and not as a binder. The brick should be dipped in a thin wash of the clay or cement and should be rubbed, shoved, and tapped into place until it touches the next brick. The thinnest joints can result only when using accurately made bricks; and the number and extent of joints are reduced by using large blocks or tile, or by using plastic refractory material to constitute the wall or cover its surface.

When the refractory brick lining is but 4½ in. thick, every fourth or fifth row should be constructed of headers (as shown at *d* in Fig. 661) so as to tie this facing to the rest of the wall. A 9-in. lining is made of headers, except that every fourth or fifth row is of stringers (as at *a* in Fig. 661). The anchoring of the lining to the outer wall must receive careful consideration.

461. Air-Cooled Refractory Furnace Walls and Arches.—(a) Walls of this type may be either entirely *self-supporting*, or they may be of the *sectionally supported construction.* In such walls the cooling air is caused to have definite flow through the hollow spaces provided in them; and if extensive areas are to be cooled this space is divided into suitable passages, or lanes, that cause the air to be distributed effectively.

(b) **Self-Supporting hollow furnace walls** may be constructed wholly of brick of the usual size; often, however, special refractory blocks or " tile " are substituted in making the lining, and special tile or metallic ties are used to provide adequate bonding between the inner and outer parts of the walls. Fig. 664 (*a*) shows one form of wall construction which, except for the bonding, is composed of standard brick—common brick for the outer position and firebrick for the lining—and in which the anchorage used between the two walls is of a form that provides for the free expansion of one wall with respect to the other when heated. In the arrangement of wall shown at (*b*) in Fig. 664, the outer wall is composed of insulating material, covered outside by a metal casing, and the inner wall is of brickwork held in place by special bonding tile which is flexibly anchored to the external steel work. As the use of standard firebrick ($9 \times 4\frac{1}{2} \times 2\frac{1}{2}$ in.) for the lining produces many joints, some engineers prefer to make the linings of large tile; one example is given in Fig. 665.



(*a*) (*b*)

Fig. 664.—Examples of Self-Supporting, Air-Cooled Refractory Furnace Walls.

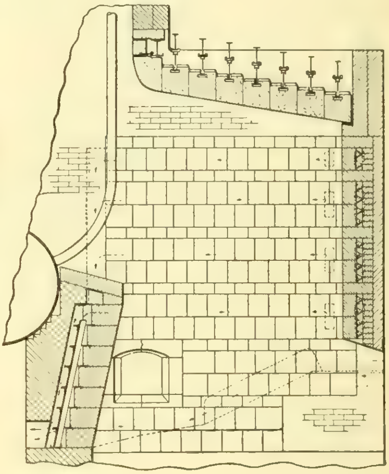


Fig. 665.—An Air-Cooled Wall Constructed of Special Large Blocks. (McLeod & Henry Co.)

(c) **Special forms of air-cooled blocks** are often used for the most severely heated parts of a wall, e.g., the surface adjacent to the fuel bed on a stoker, as in Fig. 588, and also sometimes for lining the walls completely. There are available many such blocks made of fireclay or of silicon carbide. Sometimes the blocks are provided with perforations through which air can percolate into the furnace to cool the exposed wall surface and to produce adjacent thereto a protecting oxidizing zone. Fig. 666 shows a few of many available forms (some patented) of special, air-cooled furnace blocks now on the market.[8]

[8] Each manufacturer makes a number of forms to suit various conditions; and the current designs are frequently changed to meet the rapid developments in this field.

All self-supporting walls are limited as to height by the load that can be carried by the heated lower portions; hence other forms of construction become necessary for tall walls.

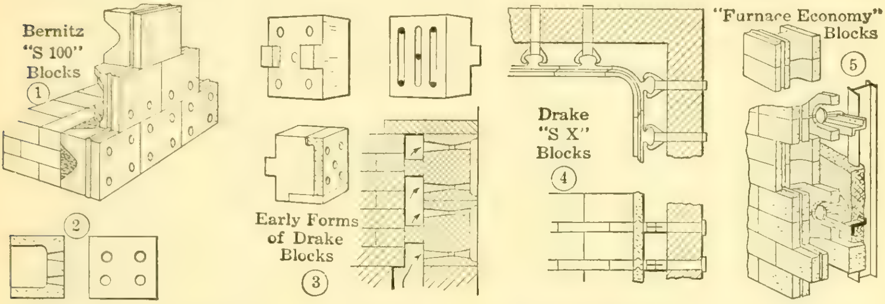


Fig. 666.—A Few Forms of Air-Cooled Furnace Blocks.

(d) For tall, hollow, firebrick walls, it is now common practice to employ some form of **sectionally supported wall construction** in which the lining, usually in 2- or 3-ft. layers, is supported on steel or cast iron brackets which are attached to an outside steel structure, thus relieving the brickwork itself from being heavily loaded. Fig. 667 shows one such arrangement. On the market are a number of patented or proprietary air-cooled walls, of which a few typical forms, each made by a different concern, are shown in Fig. 668. Most of the firms make a variety of forms, some for the walls of the boiler proper and others for special applications. The various wall constructions of this type differ as to the number of refractory shapes and metallic parts needed; the weights of the blocks, brackets and supporting steel work required; the methods of allowing for the expansion of the different parts and of sealing the joints in the lining; the initial cost; and the cost and outage-time required for making repairs.[9]

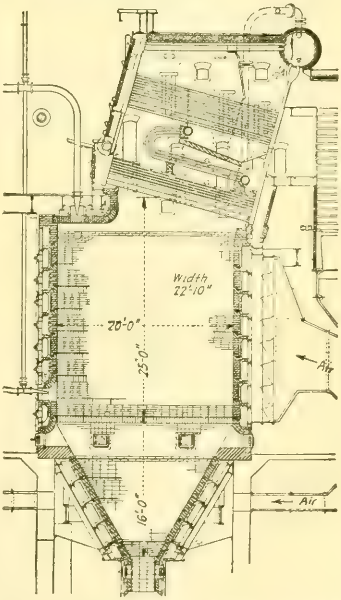


Fig. 667.—General Assembly of a Detrick Wall. (Columbia Power Co.)

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[9] For a more detailed discussion, see: "Proprietary Air-Cooled Refractory Walls," by C. S. Gladden, Trans. A.S.M.E., 1929, FSP–51–40, p. 273.

The effectiveness of the cooling of hollow walls is dependent on a number of factors which will be treated in detail in Sect. **468** (*h*).

(**e**) **Arches** over the fuel beds in furnaces were originally of the curved or **sprung type,** such as that shown in Figs. 565 and 566; but arches of this kind require great care and skill in their construction, and have end thrusts for which special provision must be made in the abutting parts supporting the arch; they also become distorted when heated, and repairs on them are difficult to make and require shutting down the unit for a long time. To avoid these objectionable features, it is now customary to use instead the **flat suspended type of arch,** examples of which are shown in Figs. 567 and 665; and a few others are given in Fig. 669. The refractory blocks in these arches are air cooled

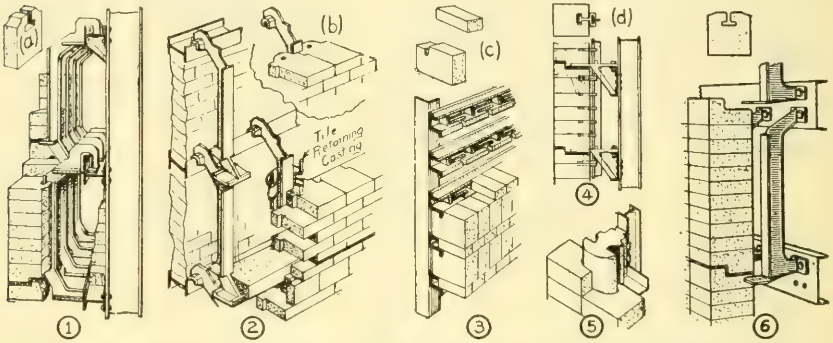

Fig. 668.—Some Forms of Sectionally Supported Wall Constructions.

(1) Detrick Co.
(2) American Arch Co.
(3) Bigelow Co.
(4) McLeod & Henry Co.
(5) DeWolf Co.
(6) Laclade–Christie Co.

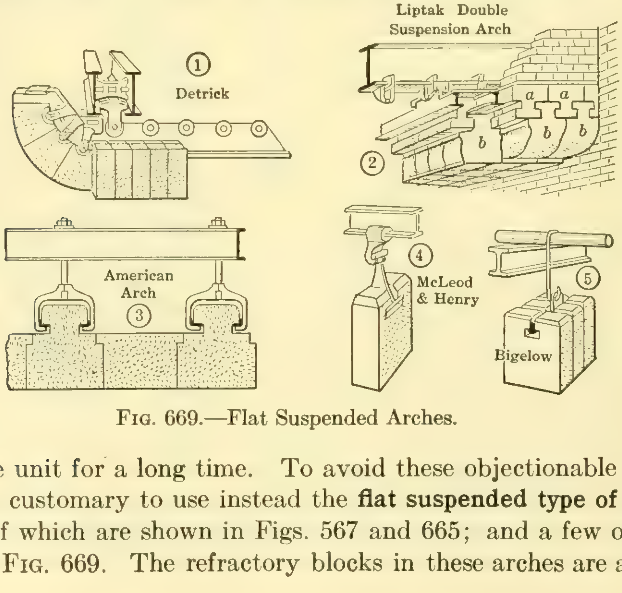

Fig. 669.—Flat Suspended Arches.

and flexibly supported, have allowance for expansion and contraction, and carry no load except their own weight. Also, the individual blocks or small groups can be easily and quickly replaced without disturbing other blocks; and all metallic parts used in the wall construction are located in positions where they will not become overheated. An arch with a surface veneer of silicon-carbide blocks which are hung by shanks or legs is shown in Fig. 670; the space for these legs is provided by merely beveling the corners of the usual refractory blocks which are to be protected. This construction is particularly useful for withstanding unusually high temperatures, rapid temperature changes, and local erosion; and, by making but slight modification to the standard blocks, it can be applied in conjunction with almost any type of suspended furnace.

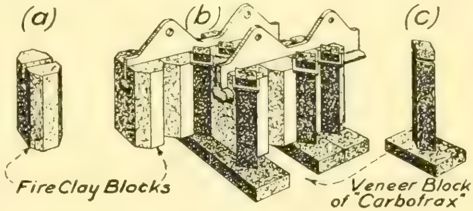

Fig. 670.—"Carbofrax" Veneered Suspended Arch. (The Carborundum Co.)

(f) **Refractory walls have inherent limitations** as to the loads, furnace temperatures, slagging actions, and mechanical abrasion that they can withstand; consequently they cannot be used economically in boiler furnaces in which service conditions cause their rapid destruction. The characteristics which limit the usefulness of refractory walls are considered in detail in Sect. **465.**

462. Water-Cooled Furnace Walls.—(a) As water-cooled walls of metal do not have, at least not to so great an extent, the limitations that are inherent in refractory wall constructions, they are to be preferred to the latter when severe conditions are to be met. However, as they are more expensive to construct than the refractory wall, they are normally substituted for them only in places where the refractory material would give trouble. For example, in a furnace fired by a multiple-retort underfeed stoker with but a fairly high degree of forcing, only the bridge wall (or back wall), which receives the greatest punishment of any, needs to be water cooled (see Fig. 588); with chain-grate stokers, when moderately operated, only the walls adjacent to the fuel bed may require protection; and with very intensive operation it may be necessary to water-cool all the walls, and the arches, if any (Figs. 575 and 581). Similarly, furnaces fired by means of powdered coal, oil, or gas may be either partly or wholly water cooled, as the conditions dictate. (See Figs. 671 and 672.)

Water walls may be classified as (*a*) *bare-plate walls*, (*b*) *bare-tube walls*, and (*c*) *covered-tube walls*. Each of these types of walls will now be discussed.

(b) Water-cooled furnaces with metallic **bare-plate walls** are inherent parts of all *internally fired boilers.* In boilers of the Scotch type (Fig. 722), the furnace is located within self-supported, water-covered, cylindrical flues; and in locomotive boilers (Fig. 721), and in some forms of vertical, internally fired, fire-tube boilers (Figs. 717, 718, and 719), the furnace is surrounded by "water legs," which are constructed of stayed plates and constitute a part of the boiler itself. Water-cooled furnaces of these kinds will not receive further consideration here. The discussion will be confined to the application of water walls to intensively operated, externally fired boilers of the water-tube type, for which the furnace walls are constructed of water-cooled tubes, which may either be bare or be protected by a metallic or refractory covering.

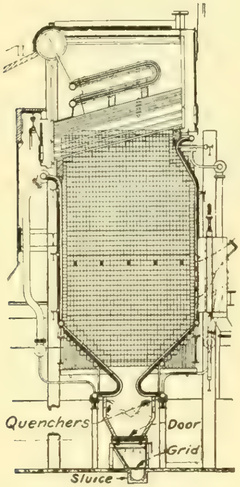

Fig. 671.—Water Wall Furnace. State Line Plant.

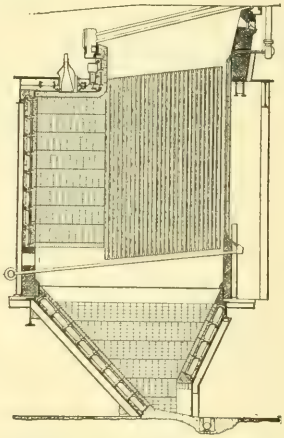

Fig. 672.—Detrick Air-Cooled Construction Combined with Water Walls and Slag Screen.

(c) A **bare-tube water wall** usually consists of closely spaced parallel tubes, the ends of which terminate in headers which are connected with other parts of the unit in such manner as to form a definite circulation system for the cooling water to traverse. Fig. 673 shows the simple elements of such a wall in which the lower header is supplied with cooling water from the overhead drum by means of a *downcomer tube* (or more briefly, a *downcomer*); and the heated, less-dense water discharged from the tubes, together with any steam entrained in it, is delivered from the upper header to the same drum by means of a *riser,* or relief tube. The water tubes shown are backed by an enclosing wall of brickwork; and the headers and downcomers are located outside of this wall so as not to be exposed to the furnace heat.

Adequate water circulation through water walls is, of course, absolutely essential. Usually the water walls, whether bare or covered, are connected into the water circulation system of the boiler proper, although a few have been constructed with independent systems of their own. Sometimes a multiplicity of downcomers and risers is used; and frequently the risers from a lower wall, such as a furnace bottom, are utilized to form the water walls for an upper portion of the furnace. Often, also, the outlet header is omitted, and the water tubes themselves, acting as risers, are carried up to a boiler drum above.

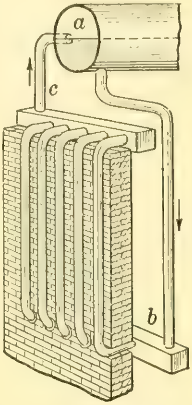

FIG. 673.—Elements of a Bare-Tube Water-Cooled Wall.

(**d**) In Fig. 674 are shown cross-sections of several typical arrangements of comparatively cheap and effective **plain bare-tube water walls,** in which the tubes are spaced a small distance apart. The tubes in *a* are located in front of an ordinary refractory wall which they partly screen and from which the rear of each tube receives reflected and radiated heat. In wall sections *b*, *c*, and *d* the back half of each tube is covered with tile or plastic refractory, a construction frequently applied in oil- and gas-fired furnaces. The refractory part of such walls is often covered on the outside with a 2- or 3-in. layer of insulating material, and in that case the wall is usually enclosed in a sheet-metal casing and the thickness of the refractory is reduced to but 9 in. In the Foster Wheeler " wing back " construction shown at *e* in the figure, the outer wall is protected, and the heat transmission to the tubes is increased, by the use of castings which are machined to give good contact with the backs of the tubes, to which each is clamped by means of a single stud that is welded to the tube.

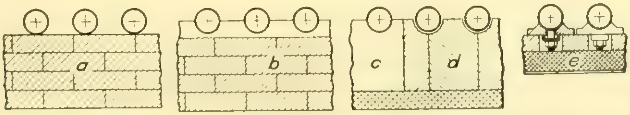

FIG. 674.—Cross-Sections of Some Bare-Tube Water Wall Arrangements.

In the arrangements shown in Figs. 675 and 676 the tubes are so closely spaced as to form substantially continuous water-cooled surfaces. In the design shown in the former, the alternate tubes enter the headers in different rows; otherwise the ligaments between the holes in the header plates would be prohibitively narrow, as they would be but $\frac{1}{4}$ in. wide. In the arrangement illustrated in Fig. 676, nearly a continuous water wall is obtained by using closely spaced bifurcated tubes of special form, in a construction which provides relatively wide ligaments between tube holes in each header, although these holes are arranged in single rows. Handhole openings, with cover plates, are placed in the water-wall headers shown in both figures, so as to give access to the interiors of the headers for expanding the tube ends into their holes, and for inspection and cleaning.

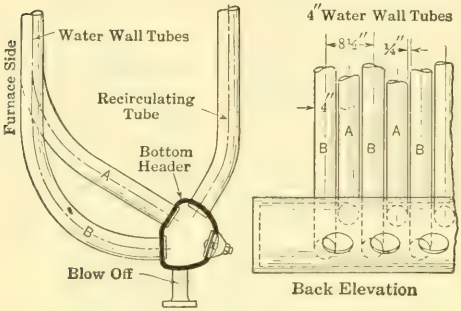

Fig. 675.—Wall with Closely Spaced Plain Tubes. (Walsh-Weidner).

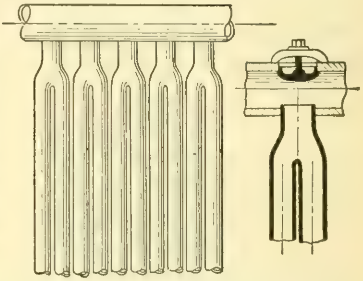

Fig. 676.—Wall with Bifurcated Tubes. (The Superheater Co.)

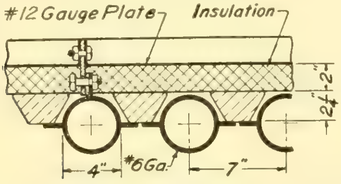

Fig. 677.—Fin-Tube Water Wall. (Combustion Engineering Corporation.)

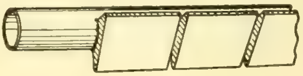

Fig. 678.—Integral-Block Water Wall. (Drake.)

(e) Another method of obtaining practically continuous water-cooled surfaces is by using the **fin-tube construction** shown in Fig. 677. In this, each tube has two longitudinal fins which are electrically welded to it diametrically opposite to each other. Ordinarily, the 4-in. tubes have fins $1\frac{7}{16}$ in. wide and the tube hole centers are spaced 7 in. apart. The tubes can be backed with a 9-in. refractory wall, or the thinner backing shown in Fig. 677 can be used. The latter is supported by lugs welded to some of the tubes.

Compared with other types of furnace walls, those constructed of thin tubes have the greatest heat-absorbing and furnace-cooling effects of any, but, if forced to high capacity, they must have the most liberal provision for the circulation of the cooling water, and this water must have a high degree of purity.

(f) In the **covered-tube type of wall,** the tubes are generally protected either by integral blocks or by attached blocks; and the latter may be entirely of metal, or of refractory, or they may be of metal with a refractory coating. The blocks are usually rectangular and flat faced, and, being placed close together, form a substantially continuous flat surface. Fig. 678 shows an **integral block construction** in which the cast-iron blocks are cast onto 4-in. tubes in a manner that causes a direct attachment thereto, thus eliminating heat-resisting air gaps between the parts. A sufficient distance is left between adjacent blocks to allow for growth of these cast-iron parts under repeated heating. Fig. 679 shows a wall construction in which cast-iron blocks, bored to gage, are shrunk onto accurately sized seamless tubes, thus closely approximating the integral construction. The tubes in armor-clad walls, such as these, range in diameter from 2 to 4 in.

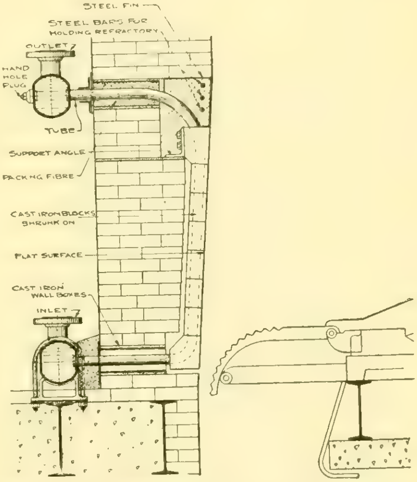

Fig. 679.—Shrunk-Block Wall. (Foster Wheeler.)

(g) **Water walls with protective metallic blocks bolted onto the tubes** are exemplified by the constructions illustrated in Figs. 680, 681 and 682. The Bailey blocks shown in Fig. 680 span between adjacent tubes to which they are clamped with intimate fit due to using ground joints and a plastic filler. The blocks may be of cast iron or steel, and the exposed faces may be bare, with smooth or ribbed surfaces, or may be covered with refractory tile or silicon carbide, the kind of block used being selected to suit its local conditions in the furnace. The refractory-faced blocks—most of which are made by placing the tile in the mold, like a core, and casting the metal on to it—are used to maintain a high temperature in the furnace, thereby aiding the completeness and rapidity of combustion. The bare blocks are used where an armored surface or much cooling is needed. The Drake " thermo-fit wall block " (Fig. 681) is of cast iron, with or without refractory facing, and is clamped to a single tube by means of two studs and a strap, an intimate joint being effected by interposing a " thermal gasket "—an embossed sheet of special alloy. In the Foster Wheeler construction shown in Fig. 682, the blocks are held in intimate contact with the tubes by means of a toggle-joint construction.

These armored walls transmit heat somewhat less rapidly than do walls of the same total width and height that are cooled by bare tubes; hence, with them, the furnace temperatures are higher under conditions that are otherwise similar; but even so, heat is transmitted less rapidly to the water in the tubes, and thus the walls do not require the use of cooling water of as great purity.[8]

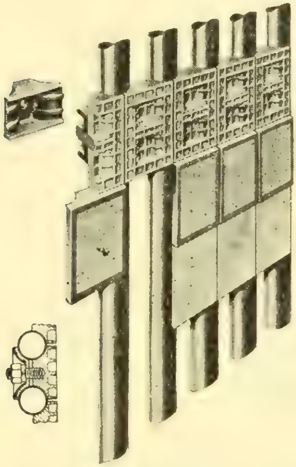

Fig. 680.—Elements of the Bailey Wall. (Block Type.)

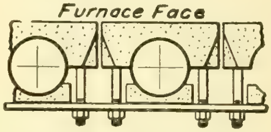

Fig. 681.—Wall with Drake Thermo-Fit Blocks.

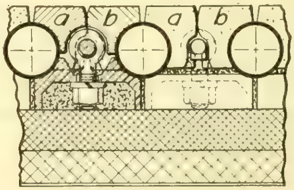

Fig. 682.—Foster Wheeler Type-B Wall.

In service, if the refractory facing used in such walls as these, and on the other refractory-protected walls to be considered next, is initially too thick, the flow of molten ash over it, as already mentioned, will wash away its surface until an equilibrium thickness is reached; and if too thin, the slag will be cooled on its surface and will build up a protective coating until an equilibrium thickness is established; thus, the combined thickness of refractory and congealed slag will vary automatically to accord with the changes in the operating conditions.

(h) **Refractory-protected water tubes** are illustrated by the construction shown in Figs. 683, 684, and 685. The Seymour wall, shown in the first of these figures, is composed of ordinary boiler tubes over which are slipped small protective refractory blocks. It is backed either by other refractory, or by insulation, or by both, and the outside is usually covered by an air-tight casing of sheet metal. The manufacturers claim that a "Seymour-Carbofrax" furnace of this construction, in which blocks of silicon carbide are used, functioned satisfactorily with release rates ranging from less than 10,000 to over 50,000 B.t.u. per hr. per cu. ft. of furnace volume.[10]

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[8] See footnote, p. 620.

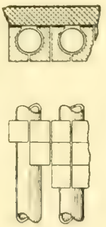

Fig. 683.—Seymour Wall. (Erie City Iron Works.)

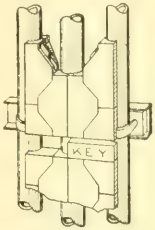

Fig. 684.—Nygaard Wall. (Bernitz Co.)

The Nygaard water-wall blocks, Fig. 684, are designed to interlock and maintain intimate contact with the abutting tubes without using clamping devices. The blocks, which are made of fireclay, "Carbofrax," or cast iron, are supported in independent sections by means of horizontal channel-bars located 6 to 10 ft. apart; and the arrangement is such that, after removing a key block, any block below can be withdrawn without disturbing any other section.

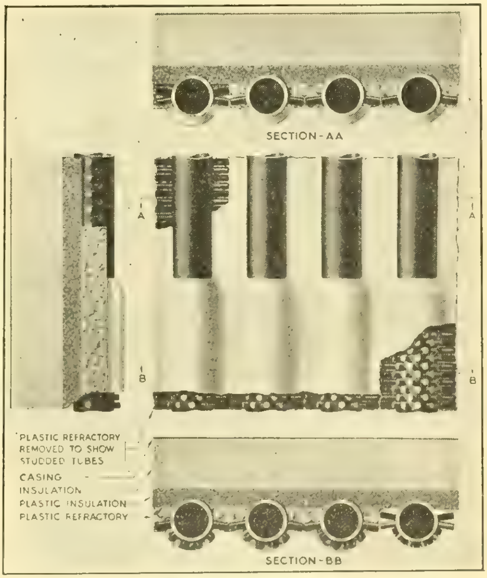

Fig. 685.—Bailey Stud-Tube Wall.

Fig. 685 shows the elements of a Bailey stud-tube wall in which iron studs are welded to the water tubes and are plastered over with refractory of suitable conductivity and thickness to give the desired rate of heat transfer and to close the spaces between the tubes. When the tubes, or parts of them, are located in the cooler regions they can be left bare, and then the studs and refractory are located only between the tubes. In the hotter regions, the studs and coating are extended to cover also all the exposed tube surface. The studs not only serve as mechanical means of support for the refractory and slag coating, but also as heat conductors from this coating to the tube wall.

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[10] Power, Nov. 20, 1928, p. 839.

The refractory covering on all water walls, such as those considered in (**g**) and (**h**) of this section, not only protects the tubes from possible blistering under high rates of operation—especially when combined with the inherent limitations in the rate of circulation and with the use of water which is not of the greatest purity—but it also enables lower rates of furnace operation to be used than are possible with bare walls.

463. Furnace Bottoms.—(*a*) The **arrangements** and **types** of furnace bottoms used in practice vary widely; and the selection for a given installation depends mainly on such considerations as the kind of fuel to be burned, the characteristics of the ash, the methods of firing and of ash removal to be used, and the cost factors involved. In furnaces operated at moderate rates by *hand or by stoker*, the ashes are usually discharged into a simple pit which is cleaned by hand. The more intensively operated stokers are provided with ash hoppers, commonly of large storage capacity (e.g., see Fig. 686), which are either lined with firebrick or have walls made up of air-cooled cast-iron plates (Fig. 588). Generally, the deposited ash is quenched by using a water spray, and it is removed from the hopper by mechanical means requiring a minimum of labor. *Oil- and gas-fired furnace bottoms* may be of solid construction or they may contain air-cooled passages. In the latter case they are commonly made of refractory hollow tile or are floored with interlocking flat tile which is supported on standard brick placed on edge.

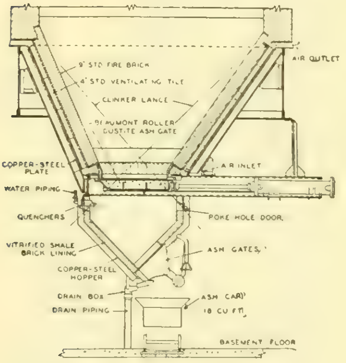

FIG. 686.—Beaumont Vestibule Ash Hopper.

(**b**) **Bottoms used with powdered-coal firing** may be very troublesome if not of suitable construction, hence must receive especially careful consideration on the part of the designer. Such bottoms are either of the *dry type* or of the *wet type.* In the former the ash is deposited in solid or congealed form, and in the latter it is maintained in a liquid state. **Dry bottoms** are the kind most commonly used, and with them it is necessary to prevent the deposited clinker from forming fused masses so large that they are difficult to remove. A dry deposit of suitable character for easy disposal is obtained (1) by placing water screens (usually 4-in. O.D. tubes spaced about 14 in. between centers) above the furnace floor, (2) by air-cooling the furnace bottom, (3) by combining (1) and (2), or (4) by water-cooling the bottom. Examples of these arrangements are given in Figs. 671 and 672, which also show hydraulic sluicing as the means for finally disposing of the slag. Fig. 686 shows the details of another air-cooled refractory bottom. This one is lined with standard ventilating tile and firebrick, and from it the ashes are dropped through a power-operated gate into a quenching hopper. The material is ultimately discharged into an ash car below.

(c) **Wet bottoms** are used with the so-called **slag-tap** or **slagging type of furnace,** Fig. 687, which has a flat hearth on which the slag—in liquid form — is allowed to form a puddle of considerable depth, and from which the accumulation is tapped off periodically, in the manner used in withdrawing iron from a cupola. By allowing a high velocity jet of water to drive the issuing stream of slag against a break-up plate, the slag is reduced to granular particles which are readily sluiced away. Such furnaces were first developed for utilizing, in powdered form, cheap coal having ash of such low fusing temperature that its use was formerly avoided. For example, in the C. R. Huntley Station in Buffalo where this type of furnace was originated, coal having about 12 to 14 per cent ash with a fusing point of 1900 to 2000 deg. fahr. was burned successfully with heat-release rates even exceeding 40,000 B.t.u. per hr. per cu. ft. of furnace and with only about 18 per cent excess air, the combustion space being surrounded by Bailey walls.

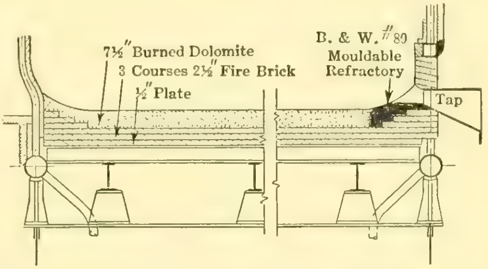

Fig. 687.—A Simple Form of Slag-Tap Furnace.

As the incandescent bottom of slag has a beneficial influence on the combustion process and on the transmission of heat by radiation, and as the flat bottom and smaller furnace volume necessary make possible reductions in the height of the steam-generating unit and in that of the building, furnaces of this type are also being developed to use coals having ash of high fusion temperature. By using horizontal turbulent burners in the lower part of the furnace, a slag having a fusion temperaure as high as 2500 or 2600 deg. fahr. may possibly be handled satisfactorily, but probably only with continuous operation under heavy loading; under reduced demands, the lower furnace temperature resulting would allow the slag to freeze and cause trouble. Greater fluidity of slag can be produced by adding a suitable flux, such as limestone, which is needed more especially during light load operation and near the tap hole.[11]

The hearth shown in Fig. 687 is fairly satisfactory when the operation is under a uniform, heavy loading and with a coal having a low-fusion ash of suitable composition. If frequent cooling takes place, this hearth will grow continuously in size, owing to the repeated formation of cracks which become filled each time with slag; and, as a result, the adjacent water walls will be displaced or the joints around the edges will be destroyed. New designs of slagging furnaces are being developed to overcome these and other difficulties.[12]

464. Water Circulation in Water Walls.—(a) As **stagnant steam** has relatively low heat conductivity, the circulation of the cooling water within a water-wall tube should under all conditions of operation be sufficiently rapid to ensure sweeping away from all parts of the exposed tube surface any films of steam that may tend to adhere thereto and thus, by blanketing this surface, prevent the cooling water itself from having intimate contact with the metal. The more sluggish the circulation, the greater will be the amount of steam formed within the tube and the greater will be the danger of tube failure from overheating.

(b) Either **natural or forced circulation** of the cooling water in a water-wall system may be used. Natural circulation, which is the kind most generally employed, is produced by having the column of fluid in the wall tubes and risers at a density lower than that of the column supplying the lower header. Rapidity of circulation in a water wall is therefore enhanced by increasing the lengths of the tubes exposed to heat in the upflow part of the circuit, and by having the downcomers and their feeders filled with liquid only and free from such exposure. The circuit should, of course, have passages of adequate size and as free as possible from resistances to flow; and the arrangement should also be such that the water is properly distributed to *all* parts of the exposed surfaces, even under the most unfavorable conditions of operation. Water walls suitable for one set of conditions will not necessarily perform satisfactorily under other conditions. Thus, such a wall designed for a high pressure unit (e.g., 1400 lb. per sq. in.), and having small tubes, so that their walls can be thin, might burn out if operated at a lower pressure (e.g., 300 lb. per sq. in.) because the volume of steam in the cooling water is much greater under the latter pressure and hence the tubes may become more or less steam-bound. Small tubes, of course, offer more resistance to flow than do large ones of the same length; and the sizes needed with extremely high pressures may be so small as even to require the use of forced circulation.

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[11] For a discussion of various fluxes and their economic use, see the progress report entitled "Fluxing of Ashes and Slags as Related to the Slagging Type of Furnace," by P. Nichols and W. T. Reid, Trans. A.S.M.E., 1932, FSP-54.

[12] For more detailed discussions, see "Evolution of the Slag-Tap Furnace at the Charles R. Huntley Station, of the Buffalo General Electric Co.," by H. M. Cushing, Trans. A.S.M.E., 1929, FSP-51-39, p. 263; "Study of Some Factors in Removal of Ash as Molten Slag from Powder-Coal Furnaces," by Sherman, Nichols and Taylor, Trans. A.S.M.E., 1929, FSP-51-51, p. 399; "Slag Tap Furnaces," by A. L. Baker, Trans. A.S.M.E., 1932, FSP-54-1, p. 1; "Slag Bottom Furnace Experiences at Hell Gate Station," by J. J. Grob, Combustion, Nov. 1934, p. 21.

(c) Flowing water naturally tends to follow paths of least resistance, which are usually the shortest ones. Hence, to obtain an **equal distribution** to all the tubes in a wall, all paths in a circuit should be of the same length or resistance. In Fig. 673 the length of passage from *b* to *c* is the same by the way of any of the intervening tubes, hence this arrangement meets the requirements; whereas, if the downcomer and riser had been connected centrally to their respective headers, the circulation through the middle tubes would be more rapid than that through the outer ones, and the latter might be subject to overheating. The distribution of circulation is, of course, also affected by inequalities in the application of heat to different tubes. In the larger walls many small risers and downcomers may be used, as in Fig. 688, and these can be spaced along the headers at such intervals as will ensure satisfactory distribution of the cooling medium. As the risers convey vapor as well as water, they should have a larger total passage area than that necessary in the downcomers. If, however, *recirculation tubes* are provided, as at *R* in Fig. 689, to return the water from the upper headers to the lower ones, the riser capacity need be only that required for handling the vapor; and the downcomers can also be reduced in passage area as they merely supply the small quantity of water needed to replace that vaporized.[13]

In Fig. 688, the risers from the slag screen are used also to form the back water wall of the furnace; and instead of discharging back into the drum from which the cooling water is received, these risers are passed through the wall of this drum and are flexibly connected to boiler tubes *ab* so as to increase the length of the heated upflowing columns.

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[13] Sometimes the back wall is only partly cooled by water-wall tubes, the rest of the cooling being accomplished by means of a radiant superheater, the elements of which are short and are located between the longer water tubes and are cooled by the main steam output which flows through them.

Fig. 690 shows another arrangement in which the water-wall tubes are carried up to an upper drum of the boiler; but in this one the extensions are located in front of the boiler tubes as shown at *a*.

(**d**) Water-cooled furnace systems are usually connected into the circulation systems of the boilers with which they function, as has been the case in all the examples so far considered. Obviously the directions of flow in the two systems should be the same in such passages as are common to both. The directions of flow in the various parts of two circuits shown in Fig. 690 are indicated by the arrows. In a few instances, the water walls have been arranged to have *independent circulation systems* of their own. An example of such an arrangement is afforded by the double water walls and slag screens shown in Fig. 691; here each element is, in effect, an independent boiler which delivers steam to the steam drum of the main boiler and has its own feedwater supply system.

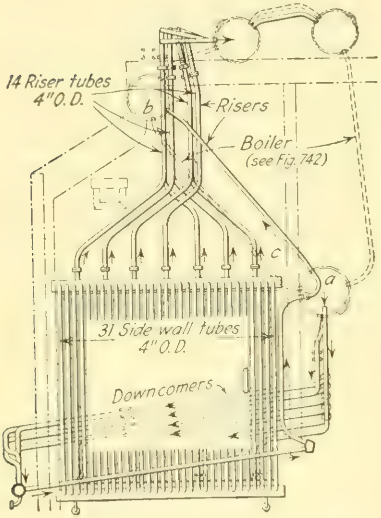

FIG. 688.—Arrangement of Water Walls and Screen which were Added to an Existing Powdered-Coal-Fired Unit to Increase Its Steaming Capacity. (Power, Apr. 21, 1931.)

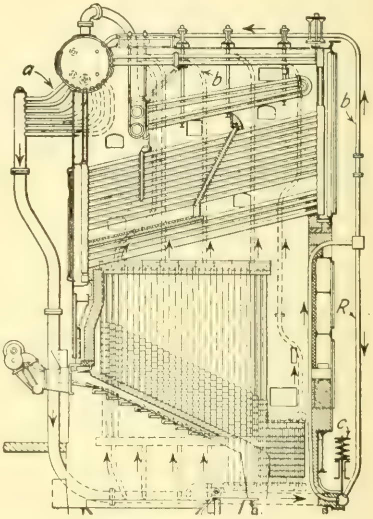

FIG. 689.—Water Walls of a Stoker-Fired Unit, Saginaw River Plant of The Consumers Power Co. (N.E.L.A. Publication No. 159.)

(**e**) In the structure of the circulation system, any possibility of having **localized stresses** occur on account of differences in expansion of the various parts must be avoided. Thus, the parts must be flexibly connected—as by means of numerous small curved tubes, such as those at *a* and *b* in Fig. 689—or flexible suspension or supports, such as the spring shown at *c* in Fig. 689, should be used. For very severe conditions of operation it may be necessary to use special alloy steels for the tubes, instead of the usual boiler-tube material.

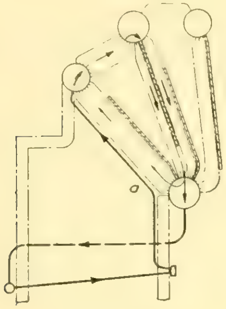

Fig. 690.

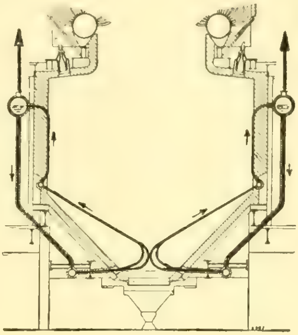

Fig. 691.

465. Refractories.—(**a**) The **desirable properties** which should be possessed by refractories are: (1) relative infusibility, (2) relatively high thermal resistance, (3) flexibility of structure, (4) low thermal expansion, (5) impermeability toward gases and liquids, (6) chemical inertness, and (7) resistance to abrasion. The usefulness of all refractories is dependent primarily upon properties (1) and (2), which are inherent in the raw materials from which the finished products are made. The remaining properties listed above may be developed to a considerable degree during the processes of manufacture; however, all these desirable characteristics are not found at the same time in any one refractory product.

(**b**) **Fireclay** is the material commonly used in making refractory brick for lining power plant furnaces. The essential constituents of all kinds of fireclay are silica (SiO_2) and alumina (Al_2O_3), and during the preparation of the bricks and other shapes these two substances react with each other and with the remaining materials present, forming one or more of a great variety of complex compounds. These compounds are formed at high temperatures, but usually well below the softening point of the mass. Some of the compounds are crystalline, some amorphous, and at any time the physical properties of the product are determined largely by the particular compounds present. Raw refractory clays fuse at temperatures between 2900 and 3200 deg. fahr., but the formation of compounds often reduces the melting point considerably.

Common fireclay refractories of good quality consist of approximately 54 per cent silica and 42 per cent alunina, together with 4 per cent of other substances. However, special clay refractories are now being made for steam-generating units, in some of which the alumina predominates. During recent years a great deal of attention has been given to the properties of refractories in service, and the special refractories which are now being developed represent an attempt to overcome the defects of the standard fireclay product. In general, high-alumina materials are more refractory than the ordinary fireclays, in the raw state.

The manufacture of fireclay refractories consists in preparing and mixing the raw materials and in molding, drying, and "burning" the product.[14] In the case of bricks which are turned out in large quantities, the molding is usually done by machinery, but other products, such as large blocks, monolithic door arches and jambs, and specially shaped baffle tiles, are often molded by hand because of their irregular shape or large size.

The important structural changes which give the finished product its physical and chemical properties take place during the burning process. The burning procedure is therefore of primary importance. Fireclay bricks and shapes are usually burned at temperatures that reach a maximum at around 2300 deg. fahr.; those burned at 2400 deg. fahr. or more are termed " hard burned," those burned at 2200 deg. fahr. or less are termed " soft burned " or " underburned."

(c) **Silicon carbide** is also used as a refractory material for the linings of boiler furnaces. It has distinctive properties which make it superior to other materials for severe operating conditions. The finished product is made in two types, the bonded and the recrystallized, in each of which the essential substance is granulated silicon carbide (silicon carbide grain).

In the manufacture of silicon carbide refractories the same general procedure is followed as that for the production of fireclay shapes. In the bonded type the silicon carbide grain, forming about 90 per cent of the product, is thoroughly mixed with ground plastic clay which serves to bind the entire mass together after firing.

Recrystallized silicon carbide refractories are made by pressing together a mass of silicon carbide grain with a small amount of an organic binder, but without the addition of clay. The molded shapes are then burned at a temperature sufficient to cause the formation of an entirely new and coherent crystal structure through the interaction of the original silicon carbide grains (which are, of course, crystalline).

For a comprehensive discussion of the composition, preparation, testing, and properties of refractories the reader is referred to the book on this subject by F. H. Norton.[15]

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[14] Powell, E. B., "Boiler-Furnace Refractories," Trans. A.S.M.E., 1925, pp. 839–948.

(d) **Failure of a refractory** may take place in one or more of the following ways: (1) by fusion, (2) by deformation under load, (3) by spalling, (4) by slag action, and (5) by changes in dimensions. Each of these kinds of failure is intimately related to the composition of the raw materials, and to the structural changes which take place within the refractories, not only when they are burned in the kiln but also during their subsequent use in the lining of the furnace. Resistance to destruction is one of the fundamental requirements of all refractories. Deterioration of the refractories in a furnace lining usually starts as soon as the lining is placed in service, and is progressive in its nature. However, if care is taken to use refractories which are suitable for the service conditions imposed upon them, deterioration will proceed at a slow rate and the lining will have a long life.

The importance of the various types of failure is so great that each one will be treated in a separate section.

(e) **Fusion** of all the refractories used in modern furnace practice takes place at temperatures around 3000 deg. fahr.; consequently it may be stated that this is not a serious cause of the failure of the material in service, since such temperatures are rarely produced at the inner surface of the lining in boiler furnaces. However, the softening and fusing points of a refractory may easily be lowered through the formation of new compounds of reduced melting point. These compounds may form from the material of the brick itself, or reactions may occur between the material of the brick and certain residues from the fuel. Fusion in service may thus be an accessory effect.[16]

(f) In general, the **crushing strength** of refractories remains nearly constant or undergoes a slow decrease as the temperature is raised, up to a particular temperature for each refractory, at which it suddenly drops to a low value.[17] For fireclay products of first quality the temperature at which this rapid decrease in crushing strength takes place is about 1800 deg. fahr.; below that temperature the material will usually show a crushing strength of 1200 lb. per sq. in., or more; above 1800 deg. fahr., the strength decreases so rapidly that at a temperature of 2500 deg. fahr. the crushing strength has dropped to about 25 lb. per sq. in. Although the mean temperature of a fireclay lining rarely, if ever, reaches 2500 deg. fahr., the inner face of the lining may easily reach this, or a higher temperature. The allowable unit load which may be safely superimposed upon the refractories near the bottom of a furnace wall must therefore be kept fairly low, certainly below 25 lb. per sq. in., if the possibility of deformation under load is to be eliminated.

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[15] Norton, F. H., "Refractories," McGraw-Hill Book Co., 1931.

[16] Sherman, R. A., "A Study of Refractories Service Conditions in Boiler Furnaces," United States Bureau of Mines Bull. 334, 1931.

[17] Trinks, W., "Industrial Furnaces," Vol. 1, p. 188, John Wiley & Sons, 1923.

(g) **Spalling** has now been tentatively defined by the American Society for Testing Materials[18] as any breaking or cracking of bricks to expose fresh surface, whether due to thermal shock, to pinching because of expansion, to changes in the structure of the brick because of absorption of slag, or to forces from the fireman's tools. Considerable research work has been done, both in this country and abroad, in an endeavor to discover the true mechanism of spalling. This work really consists in studies of the stresses and fractures in brittle materials. In spite of the progress which has been made, much work remains to be done.

The severity of thermal shock is closely related to the rate of temperature change of the inner face of the brick when the furnace is heated up or cooled down. In furnaces fired with oil, gas, or powdered coal, this rate of temperature change may reach high values, owing to the rapid variations which can be produced in the rate of combustion. In an oil-fired furnace the temperature of the inner face of a solid firebrick wall may rise as much as 2000 deg. fahr. per hr., and the same wall may cool down at the rate of 800 to 1000 deg. per hr. This is probably the most severe type of service which fireclay refractories are expected to withstand. However, the rate of temperature change of the inner face of solid firebrick walls in both powdered coal and stoker-fired furnaces may also be well over 1000 deg. fahr. per hr. during heating, and about half that figure during cooling if the load has dropped sharply from a high value.

Sherman[19] states that spalling is worst in furnaces subjected to variable loads, and that the tendency to spall increases with an increase in the size of the individual refractory shapes.

The use of silicon carbide refractories is one of the most feasible means of preventing a furnace wall from spalling. Since the conductivity of silicon carbide refractories is from 6 to 10 times as much as that of fireclay products, the former material stands sudden temperature changes very well. The temperature gradient through a given thickness of silicon carbide lining is much less than that through the same thickness of fireclay lining when the two are similarly installed and subjected to the same furnace conditions, hence any changes in the temperature at the inner surface will, with the silicon carbide material, be quickly transmitted to the interior of the lining.

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[18] "Tentative Definitions of Terms Relating to Refractories," Proc. A.S.T.M., v. 27, pt. 1, 1927, p. 839.

[19] See footnote, page 637.

(**h**) **Slag action** is one of the greatest sources of trouble with the fireclay refractories used in furnaces. This action results in the erosion of the wall due to the fluxing of portions of the refractory by the ash of the solid or liquid fuel, or to the building up of layers of solidified slag upon the wall surface.

Slag erosion is initiated by the impingement of molten or semimolten particles of ash upon the surface of the refractory. Although the refractory itself may be below its softening point, the reactions which take place between the fluxing constituents of the ash and the refractory result in the formation of new compounds of reduced melting point at the surface of the lining. The resulting slag, now in the molten state, flows by gravity over the surface of the lining, penetrating to some depth because of the porosity of the firebrick, and continuing the erosion process.

In the case of **slag adhesion,** the compounds formed by the interaction of the ash and the refractory are not fluid at the existing furnace temperature, consequently the resulting slag builds up upon the surface of the lining. This forms a protection for the refractory, but when it has grown to the point at which it interferes with the firing of the furnace, its removal always results in some damage to the lining.

Slag erosion is worst when the coal used contains an ash having a low fusing temperature. As far as slag adhesion is concerned, it may be stated that the ash from practically any coal will at least become plastic and adhere to the firebrick at some temperature obtainable during normal furnace operation. By some authorities the nature of the atmosphere in the neighborhood of the refractories is considered important. A reducing atmosphere is believed to facilitate the reactions between the ash and the refractory.

Unfortunately, high slagging resistance and high spalling resistance are not usually found in the same fireclay product, since the flexible structure which is necessary to resist spalling is composed of constituents which allow considerable slag penetration and the formation of many easily melted compounds. Among the refractories in common use, silicon carbide combines, to a high degree, the favorable characteristics for resisting both slagging and spalling. Its spalling characteristics have already been discussed, and, as far as slagging is concerned, this refractory is practically inert except in the presence of an ash high in iron when exposed to an oxidizing atmosphere. In this case ferrosilicates are formed and the refractory disintegrates.

An exceedingly favorable combination of high slagging resistance and high spalling resistance is found in a clay refractory which is now being developed. This product is composed entirely of mullite, the most stable compound of silica and alumina, having the chemical formula $3Al_2O_3 \cdot 2SiO_2$. Mullite shows practically no softening below its melting point of 3290 deg. fahr., and in addition to being non-spalling and resistant to slag action, undergoes almost no permanent volume change while in service.

(i) **Changes in the dimensions** of a refractory brick when it is subjected to furnace temperature may be either temporary or permanent. The nature and extent of temporary changes depend upon the coefficient of expansion of the material; permanent changes result from the formation of new constituents within the structure at the temperatures to which the material is subjected. Reasonable freedom from permanent changes, for a fireclay product, can be achieved only by burning the material at a higher temperature than any to which it will be subjected in service. On the other hand, such materials as silicon carbide and mullite have naturally stable structures within the range of temperatures possible in furnace operation.

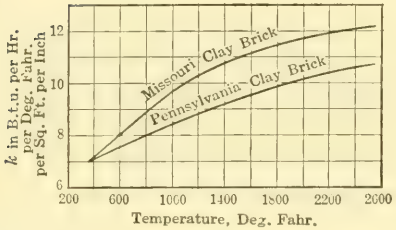

FIG. 692.—Specific Thermal Conductivity (*k*) of Fireclay Brick.

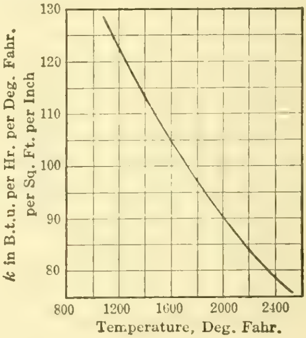

FIG. 693.—Specific Thermal Conductivity (*k*) of Silicon Carbide Brick, Recrystallized Type.

(j) The **conductivity of a refractory** is its most significant thermal characteristic. This property is expressed in terms of the coefficient of conductivity which, for a given material, varies over a particular range of values as the mean temperature of the refractory is changed.

The variation of the coefficient of conductivity with the mean temperature of the material for two kinds of fireclay refractories, as well as for silicon carbide of the recrystallized type, is shown by the curves of Figs. 692 and 693, from Kolb.[20]

[20] Thesis, "Heat Transfer in Air-Cooled Furnace Walls," by Robert P. Kolb, Cornell University, June, 1932.

Note that the conductivity of fireclay brick increases with an increase in the temperature; and this kind of variation is common to almost all the commercial refractories, the notable exceptions being magnesite (seldom used in heat-power work) and recrystallized silicon carbide.

Although high thermal resistance is one of the functions of most refractories, there are cases in which the refractory is expected to serve as a medium for the controlled transfer of heat to air passages or water tubes. In addition, the ability of the material to transfer heat may be of considerable value in lengthening the life of the refractory itself.

(k) The **utility of a refractory** is determined by all the properties previously treated. A chart showing the relation between the properties of refractories and the conditions of service to which they are subjected, has been devised by S. M. Phelps [21] and is given on page 642.

466. Insulation.—(a) Some form of **thermal insulation** is often incorporated in the structure of furnaces, flues, and other equipment used for the generation and utilization of heat, to reduce the rate of heat transmission through these structures to the outside air. The application of a layer of insulation against the outer face of a refractory furnace lining causes a reduction in the temperature of the exposed surface of the wall structure and thus decreases the rate of heat transmission to the air in the room and to nearby objects " seen " by the wall.

Insulating materials suitable for application to the equipment mentioned above are known as high-temperature insulators, and are of two kinds: (*a*) refractory insulators which may be used in direct contact with the furnace gases, and (*b*) insulators which must be installed behind separate and more refractory inner linings. All high-temperature insulators should possess the following qualifications: (1) low conductivity, (2) mechanical strength sufficient to withstand shipping, handling, and application, (3) low shrinkage when heated, (4) resistance to the decomposing effect of high temperatures, (5) resistance to the effects of moisture, and (6) light weight. The maximum temperature which an insulator can withstand continuously without breaking down determines its suitability for a given installation; second in importance is low thermal conductivity. In general, the characteristics of insulating materials are such that, as the ability to withstand high temperatures increases, the conductivity also increases.

All high-temperature insulators owe their comparatively low conductivity to the presence, within their structure, of a large number of small air cells which effectively reduce the overall transfer of heat through the material by conduction. It is essential, however, that the

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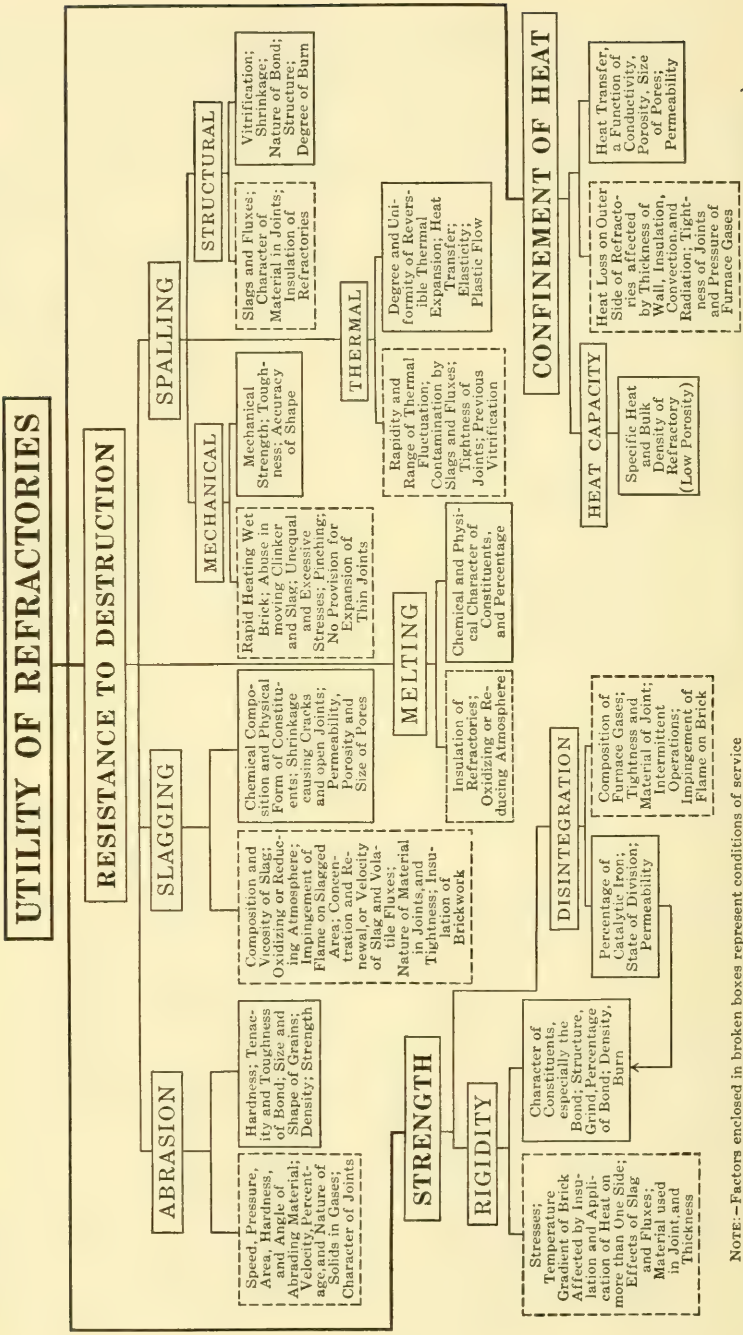
[21] See "Properties of Refractories and Their Relation to Conditions in Service," Trans. A.S.M.E., 1928, FSP-50-8, p. 52.


UTILITY OF REFRACTORIES
RESISTANCE TO DESTRUCTION
ABRASION
Speed, Pressure, Area, Hardness, and Angle of Abrading Material; Velocity, Percentage, and Nature of Solids in Gases; Character of Joints
Hardness; Tenacity and Toughness of Bond; Size and Shape of Grains; Density; Strength
SLAGGING
Composition and Vicosity of Slag; Oxidizing or Reducing Atmosphere; Impingement of Flame on Slagged Area; Concentration and Renewal, or Velocity of Slag and Volatile Fluxes; Nature of Material in Joints, and Tightness; Insulation of Brickwork
Chemical Composition and Physical Form of Constituents; Shrinkage causing Cracks and open Joints; Permeability, Porosity and Size of Pores
MELTING
Insulation of Refractories; Oxidizing or Reducing Atmosphere
Chemical and Physical Character of Constituents, and Percentage
SPALLING
MECHANICAL
Rapid Heating Wet Brick; Abuse in moving Clinker and Slag; Unequal and Excessive Stresses; Pinching; No Provision for Expansion of Thin Joints
Mechanical Strength; Toughness; Accuracy of Shape
THERMAL
Rapidity and Range of Thermal Fluctuation; Contamination by Slags and Fluxes; Tightness of Joints; Previous Vitrification
Degree and Uniformity of Reversible Thermal Expansion; Heat Transfer; Elasticity; Plastic Flow
STRUCTURAL
Slags and Fluxes; Character of Material in Joints; Insulation of Refractories
Vitrification; Shrinkage; Nature of Bond; Structure; Degree of Burn
STRENGTH
RIGIDITY
Stresses; Temperature Gradient of Brick Affected by Insulation and Application of Heat on more than One Side; Effects of Slag and Fluxes; Material used in Joint, and Thickness
Character of Constituents, especially the Bond; Structure, Grind, Percentage of Bond; Density, Burn
DISINTEGRATION
Percentage of Catalytic Iron; State of Division; Permeability
Composition of Furnace Gases; Tightness and Material of Joint; Intermittent Operations; Impingement of Flame on Brick
CONFINEMENT OF HEAT
HEAT CAPACITY
Specific Heat and Bulk Density of Refractory (Low Porosity)
Heat Loss on Outer Side of Refractories affected by Thickness of Wall, Insulation, Convection, and Radiation; Tightness of Joints and Pressure of Furnace Gases
Heat Transfer, a Function of Conductivity, Porosity, Size of Pores; Permeability

(*American Refractories Institute Fellowship, Mellon Institute, March, 1927.*)

NOTE:—Factors enclosed in broken boxes represent conditions of service while those in solid boxes refer to properties of refractories.

cells be as small as possible; if they are, roughly, large enough to be seen with the naked eye, the transmission of heat across them can take place readily, in most cases, both by radiation and convection, and the material has an increased conductivity. On the other hand, the more dense materials, which are best able to withstand high temperatures, contain a comparatively small number of air cells and are thus capable of transmitting a considerable amount of heat by conduction, with a consequent increased conductivity.

Four basic materials are used at the present time in the manufacture of high-temperature insulators. These are (1) asbestos fiber, (2) diatomaceous or infusorial earth, (3) alumina, and (4) kaolin. Several of these materials are naturally cellular, but for those that lack this property, a certain amount of porosity is developed in the finished product during manufacture.

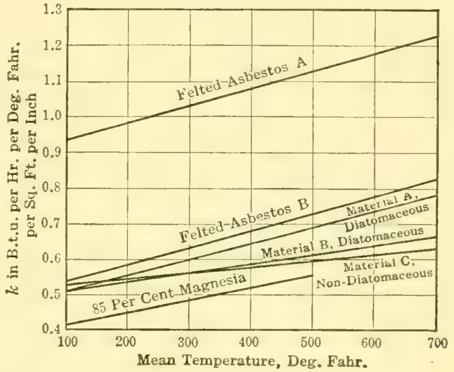


FIG. 694.—Specific Thermal Conductivity (*k*) of high-temperature Insulation compared with 85 per cent Magnesia.

(b) **Asbestos fiber** is produced from the raw mineral asbestos by successive crushings. This material contains a large number of small air spaces and is therefore a good insulator. It holds considerable water of crystallization and possesses the property of retaining its physical strength until this water is given off. This property prevents the use of asbestos fiber alone at temperatures above 1000 deg. fahr. For use as a high-temperature insulator the fiber is felted and pressed into blocks or sheets having high mechanical strength. Owing to the fact that the pressing operation reduces the effectiveness of the air spaces, the thermal conductivity of this product is not as low as that of other insulators in which the many cells are retained. The conductivity curves, as given by Zeiner,[22] for two kinds of felted asbestos insulation are shown in Fig. 694. The difference in the conductivities of the two felted-asbestos products is due to the characteristics of the raw materials.

Asbestos fiber is often mixed with raw or lightly calcined infusorial earth, or with carbonate of magnesia, the resulting product being molded and used as high-temperature insulation. The latter products will withstand temperatures up to 1550 deg. fahr. and at a mean temperature of 750 deg. fahr. they have, according to Norton,[23] an average value of k equal to about 0.78.

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[22] See Power, Jan. 20, 1931, p. 104.

(c) **Diatomaceous or infusorial earth** is, at the present time, the most widely used high-temperature insulator. This material is pure silica and is composed of the remains of the simplest form of organisms, called " diatoms." Although the organic matter has entirely disappeared, the microscopic cells are left, to the number of many millions in each cubic inch of the mineral.

Diatomaceous earth, in the raw state, will withstand temperatures up to 1600 deg. fahr, and a large amount of it is used for insulation in this form, the preparation consisting merely in cutting the material into bricks which are dried before being used. Several representative samples of this raw product had thermal conductivities as shown by the curves for Material A and Material B in Fig. 694, and by the curve for Natural Diatomite in Fig. 695, also from Zeiner. Blocks made from the natural material show considerable brittleness, hence the raw or lightly calcined earth is often ground, mixed with asbestos fiber, and molded, in order to improve its mechanical strength. The thermal properties of the latter product are about the same as those of the raw material alone.

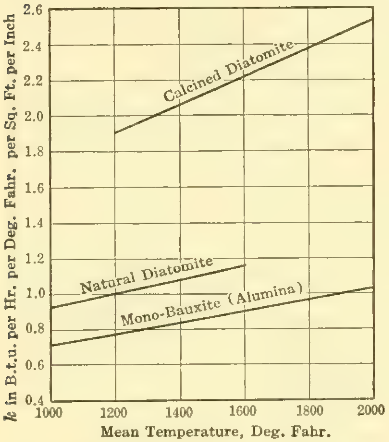

Fig. 695.—Specific Thermal Conductivity (k) of Furnace Insulation and Firebrick.

The refractoriness of infusorial earth may be increased considerably by calcination. Several insulators have a base of moderately calcined earth to which is added organic material and perhaps a certain amount of clay. During the subsequent firing of the product the organic substance is burned away and the porosity of the bricks is thus increased. This tends to offset the increased density of the material and thus prevents a serious rise in the conductivity. Bricks made in this way can be used for temperatures up to 2000 deg. fahr., and have an average value of k equal to about 1.1 at a mean temperature of 750 deg. fahr. A product containing more highly calcined earth, and fired at a higher temperature, has a somewhat higher conductivity and will withstand a maximum temperature of 2500 deg. fahr. Owing to the variation in structure and composition of this class of insulation as produced by different manufacturers, all the products of this type have conductivities which follow approximately the curve for Calcined Diatomite shown in Fig. 695.

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[23] See footnote, page 637.

(d) **Alumina** forms the base of an insulating material composed of mono-bauxite ($Al_2O_3 \cdot 2H_2O$), asbestos fiber, and clay. This product can be used for temperatures up to 2000 deg. fahr. and has high mechanical strength. Its thermal conductivity, as shown by the curve in Fig. 695, is considerably less than that of calcined infusorial earth which is prepared for similar service, and is even less than that of natural diatomite. The alumina-base product derives its favorable properties largely from the presence of the cellular asbestos fiber.

(e) **Refractory insulators** must possess the favorable characteristics of both a refractory and an insulator. They must resist spalling and shrinkage like a high-quality firebrick and should be able to withstand temperatures up to 3000 deg. fahr. These requirements eliminate from consideration all the materials described above, and have made necessary the development of an entirely new class of product.

(f) **Kaolin** ($Al_2O_3 \cdot 2SiO_2 \cdot 2H_2O$) is used to a considerable extent as a material for refractory insulators. Since pure kaolin fuses at 3245 deg. fahr., products made from it can be used safely at a maximum temperature of 3000 deg. fahr. A porous structure is developed during the process of manufacture and gives the insulator a low thermal conductivity as well as a good resistance to spalling. However, the resistance to slag action is poor, and this limits the use of the material to furnaces in which there is no residue from the fuel. One particular refractory insulator has been described by Norton, who states that the specific thermal conductivity of this product rises from a value of 1.45 at a mean temperature of about 400 deg. fahr. to a value of 2.9 at a mean temperature of about 2000 deg. fahr. These values are very nearly the same as those for calcined infusorial earth, and the value for 2000 deg. fahr. is about 25 per cent of that of first-quality firebrick.

467. Reception and Transmission of Energy in a Furnace.—(a) The various sources of the heat supplied to a furnace, and the transmission or disposal of this heat, usefully or otherwise, are shown diagrammatically in the chart given in Fig. 696, which is largely self-explanatory. The furnace to which the chart applies is assumed to have a combination of air-cooled and water-cooled walls, and the unit includes an air preheater. Of the energy supplied to the furnace, a part is transferred to the furnace walls—mainly (*a*) by radiation from solid masses of fuel and slag, (*b*) by radiation from the gases and fly-ash, and (*c*) by convection—and the remainder is delivered to the boiler or retained by the exit gases. Part of the heat received by each element of area of the refractory wall is radiated and reflected to colder surfaces seen by the element; thus, the surface temperature of the wall at any point has a value dependent upon the temperatures and relative extents and positions of the other refractory and water-cooled surfaces within the furnace. The water-cooled surfaces are called, collectively, the *Cold-Surface* (C.S.), and usually include not only the water-wall and slag-screen surfaces but also any boiler-tube surface that is exposed to the furnace. The relative proportion of the wall surface that is water cooled (including refractory-

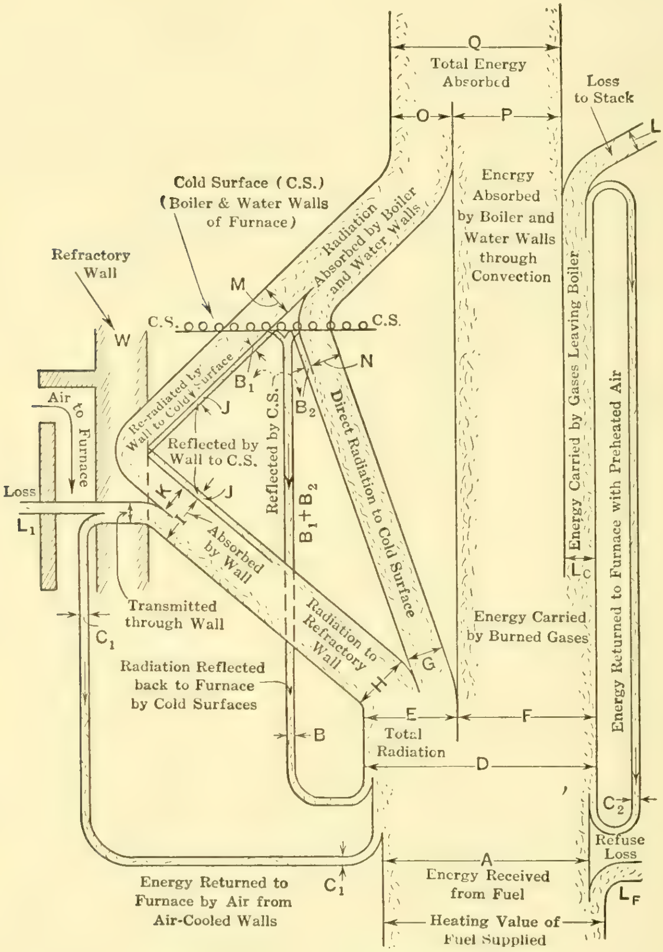
Q
Total Energy Absorbed
O
P
Loss to Stack
L₂
Cold Surface (C.S.) (Boiler & Water Walls of Furnace)
Radiation Absorbed by Boiler and Water Walls
Energy Absorbed by Boiler and Water Walls through Convection
Refractory Wall
M
C.S.
C.S.
W
N
Re-radiated by Wall to Cold Surface
B₁
J
B₂
Energy Carried by Gases Leaving Boiler
Energy Returned to Furnace with Preheated Air
Air to Furnace
Reflected by Wall to C.S.
Reflected by C.S.
Direct Radiation to Cold Surface
J
B₁+B₂
Loss
L₁
K
I
Absorbed by Wall
L_C
Transmitted through Wall
Radiation to Refractory Wall
Energy Carried by Burned Gases
C₁
G
H
Radiation Reflected back to Furnace by Cold Surfaces
E
F
B
Total Radiation
D
C₂
Refuse Loss
A
Energy Returned to Furnace by Air from Air-Cooled Walls
C₁
Energy Received from Fuel
L_F
Heating Value of Fuel Supplied

FIG. 696.—Energy Streams for a Furnace.

faced water walls) has an important bearing on the temperature obtained within the furnace. Cold surfaces can be substituted for refractory ones in a wall to limit the highest furnace temperature obtained under the heaviest loading to a value satisfactory with respect to the fusibility of the ash and to the properties of the refractory material used in the walls; but such substitution may increase the difficulties of operation under light loads. The proportion of cold surface, or " fraction cold," to be used in a new installation should, therefore, be selected with care so as always to obtain a satisfactory state within the furnace under the full range of the loading and other operating conditions.

(**b**) The **fraction cold** (ψ) is defined as

$$\psi = \frac{\text{actual extent of cold surface in the furnace}}{\text{maximum possible extent of cold surface in furnace}}$$

using for the surfaces, unless otherwise stated, the projected areas of the cooling elements, and measuring them with respect to the wall faces at which they are placed. A completely water-cooled cubical furnace, with roof fully covered by boiler tubes, would have five " cold " walls if stoker fired, and six such walls (including the bottom) if fired by powdered coal, oil, or gas. Then, if the actual cubical furnace has n water-cooled walls, including the roof, the value of ψ would be $n/5$ in the former case, and $n/6$ in the latter. Fig. 697 shows in a general way the proportion of the energy released within the furnace that is absorbed by the furnace walls having various values of the fraction cold, the remainder of the energy being, of course, transmitted by convection to the boiler tubes and other elements of the unit beyond the furnace; and Fig. 698 illustrates[24] the effect of load variation on the rate of total energy release, on the relative proportions of energy absorbed by the water walls and by the rest of the unit, and on the rate of heat transfer per square foot to the furnace wall of a unit having a furnace with $\psi = 1$. (The unit to which these curves of Fig. 698 apply is that shown in Fig. 545, and test data for it are given on pages 454 and 455.)

(**c**) The rate of total energy release needed within a furnace to meet the load which a proposed steam-generating unit is to carry can be readily determined; and, after selecting what appears to be a suitable specific energy release rate for the proposed wall construction and conditions, the **tentative size** of the furnace can be computed. The rates of heat transmission to and from the walls, and the furnace and wall

[24] From "Comparative Performance of a Pulverized-Coal-Fired Boiler using Bin System and Unit System of Firing," by A. E. Grunert, Trans. A.S.M.E., 1931, FSP–53–4, p. 43.

temperatures, can then be investigated for the proposed type of wall, and finally the design may be modified on the basis of the information thus obtained.

The probable rate of heat transfer to the furnace walls can be determined by applying the principles already given in Chapter XXVI (Heat Transmission). However, in making the analysis, the transmission of heat to the walls by conduction and by convection may ordinarily be neglected without serious error, as the former is usually very small and the latter is of the order of only about 5 per cent of the total energy, a percentage well within the errors involved in many of the assumptions that must be made; hence, for most purposes, only the transmission of the energy by *radiation* need be considered. The method of determining the rate of heat transfer to a furnace wall by the latter method is illustrated in the numerical example given on page 280 for a furnace fired by pulverized coal. A more elaborate study of such analytical determinations is contained in the writings of Wohlenberg and others, to which reference will be made later.

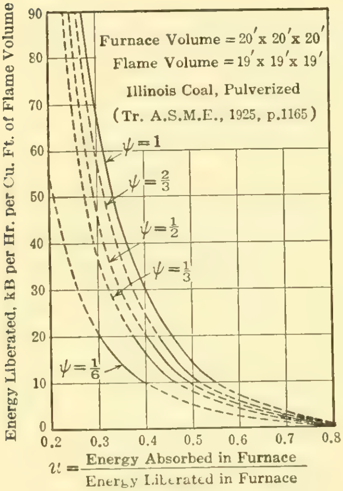

FIG. 697.—Influence of Fraction Cold on the Heat Absorbed by Furnace Walls. ($kB = 1000$ B.t.u.)

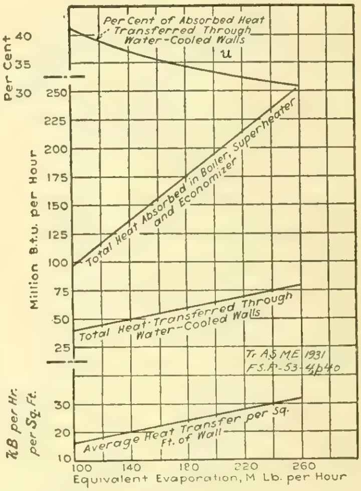

FIG. 698.—Typical Transference of Heat to the Furnace Walls and Other Parts of the Unit Shown in Fig. 545.

(d) When the furnace walls are **part refractory and part water cooled** it is necessary to consider not only the heat transmission from the flame to the respective wall surfaces but also the interchange between the refractory surfaces and the colder ones. The proportion of the total heat transmitted to the walls in such a case is intermediate between that obtained with water-cooled walls alone and that when all the walls are refractory ones. Analytical methods of treating this case are given by Haslem and Hottel[25] and by Wohlenberg and Lindseth,[26] and are somewhat involved.

The authors last mentioned have proposed, however, a method of simplifying the analysis by the use of certain charts which show clearly the influence of the fraction cold. The first step in applying this method is to determine the fraction (u_c) of the furnace energy liberated that would be absorbed by the walls of a completely water-cooled furnace (i.e., one with $\psi = 1$) of the same size as the furnace under consideration. This may be approximated from a chart such as the typical one given in Fig. 697. Then the fraction (u) absorbed by the water walls of the actual furnace under investigation is obtained by multiplying u_c by a suitable factor γ based upon the actual fraction cold. This factor, which Wohlenberg terms the "coefficient of emission reference factor," is defined as

$$\gamma = \frac{\text{energy absorbed by water walls in given furnace cavity}}{\text{energy absorbed in same cavity with } \psi = 1}.$$

Typical relations between γ and ψ are given in Fig. 699, taken from the last reference. Thus, for any value of ψ and the corresponding value of γ, the fraction of available energy absorbed by the water-wall portion of the furnace is $u = \gamma u_c$.

The curve in Fig. 699 shows that γ does not increase in direct proportion to the increase in ψ, and that the mean temperature of the furnace gases is influenced far less by an increase in ψ from 0.3 to 1.0 than it is by an increase from 0 to 0.3. Thus, for each set of conditions, there is a certain fraction cold, ψ, that will require for a desired final gas temperature the least amount of total surface in the furnace and boiler, and this value of ψ is the most effective fraction cold for these conditions. (The curve passes through the origin because the refractory part of the wall is assumed to be solid and completely insulated and thus does not transmit heat away from the furnace.)

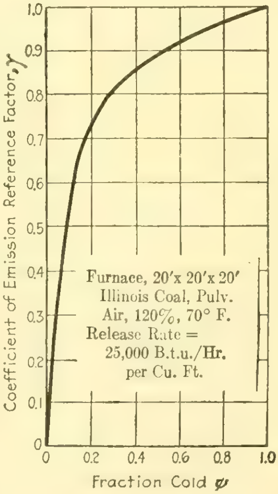

FIG. 699.

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[25] "Combustion and Heat Transfer," Trans. A.S.M.E., 1928, FSP-50-3, p. 9.

[26] "The Influence of Radiation in Coal-Fired Furnaces on Boiler-Surface Requirements, and a Simplified Method for its Calculation," Trans. A.S.M.E., 1926, p. 849.

One example of the determination of the most effective value of ψ, in this case, to reduce the gas temperature to 2600 deg. fahr., is given in Fig. 700, in which the total surface shown by the ordinates includes, in addition to the water-cooled surface facing the furnace, the amount of convection surface within the boiler that must be passed over by the gases before their temperature is reduced to the desired value.[27]

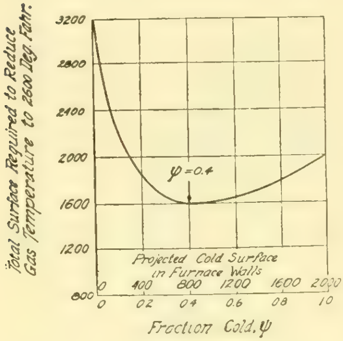

FIG. 700.—Influence of Fraction Cold on Total Surface Requirement.

(Furnace No. 2, 8000 cu. ft.; 80,000 B.t.u. per cu. ft. per hr.; Illinois coal; excess air, 40 per cent; stoker fired.)

Discussion of the methods of computing the data needed for constructing charts such as those just shown is beyond the scope of this book. For a detailed study of them consult the references given in this section and Sect. **469.**

468. Heat Transmission through Furnace Walls.—(a) **The chief factors** which affect the transmission of heat through any furnace wall are (1) the nature of the materials composing the wall, (2) the relative location of the various elements of the wall structure, and (3) the method of cooling employed.

(b) **Solid walls transmit heat by conduction,** the quantity transferred varying inversely with the wall thickness and directly with the conductivity of the material or combination of materials of which the wall is built. Whereas refractory linings with thicknesses up to 18 in. were commonly used in the earlier designs of solid walls to reduce the rate of heat transfer, a layer of insulation behind a 9-in. lining is now depended upon to produce an even more favorable result than that formerly obtained with the refractory alone.

The effect of the inclusion of insulation in a solid-wall structure is illustrated by the diagrams in Fig. 701. Comparing diagrams *B*, *C*, and *D*, with diagram *A*, it should be noted that the addition of more and more insulation causes a progressive reduction in the temperature of the exposed outer surface of the wall. This fact is of great importance because, under steady operating conditions, the rate of heat transmission through the wall just equals that from the exposed surface to the outside air, and the amount of this transmitted heat is therefore directly dependent upon the temperature of the exposed surface.

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[27] "Heat Reaction in Boiler Furnaces," by W. J. Wohlenberg, Trans. A.S.M.E., 1930, FSP-52-26, p. 177.

The presence of thermal insulation in a solid-wall structure also causes a reduced temperature gradient through the refractory lining. If the furnace temperature is not excessive, this will have a beneficial effect upon the lining because the refractory is thus at a more uniform temperature throughout and the tendency toward spalling is diminished. However, if so much insulation is used that the gradient is quite small, and if at the same time very high temperatures prevail in the furnace, the lining is likely to suffer considerably through softening or incipient fusion as well as slag action.

(c) **Heat transfer data for solid-wall structures** composed of various combinations of firebrick, insulation, and red brick, are shown in Table LXI. The relative valuesof different arrangements of firebrick and insulation, as compared with plain firebrick, are also indicated in Figs. 702 and 703, in which it may be seen, for example, that a wall consisting of 9 in. of firebrick backed by $2\frac{1}{2}$ in. of insulation has the same heat loss and the same outer-surface temperature (for a given inner-surface temperature) as a solid firebrick wall 18 in. thick. It is evident that many different combinations of elements may be used in the construction of a solid furnace wall; hence the kind of materials to be used, together with their amount and disposition within the structure, must be decided on a purely economic basis, taking into account all the

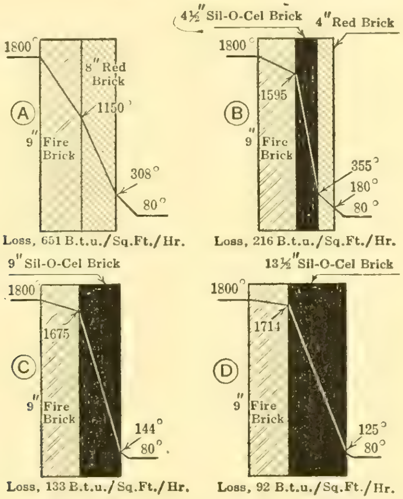

FIG. 701.—Temperature Gradients and Heat Losses through Uninsulated and Insulated Walls.[28]

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[28] Reproduced by permission from "High Temperature Insulation," published by Johns-Manville Corporation.

factors connected with the generation and utilization of the heat in the furnace.

TABLE LXI

Heat Losses through Solid Furnace Walls *

B.t.u. per Hr. per Sq. Ft.

Room Temperature = 80° F.

Thicknesses, Inches			Hot-Surface Temperatures, Deg. Fahr.						
Firebrick	Insulation	Red Brick	1000	1200	1400	1600	1800	2000	2500
13½	0	8	240	305	375	445	525	605	835
9	0	8	295	370	455	545	645	745	1015
9	2½ *A*	0	170	217	264	312	365	417	
9	4½ *A*	0	110	138	169	203	237		
9	4½ *A*	8	95	118	143	170	198	228	310
9	4½ *B*	0	215	267	325	383	450	515	700
9	4½ *B*+4½ *A*	0	....	....	....	150	175	200	275
9	4½ *C*	0	....	....	....	390	455	520	700
9	4½ *C*+4½ *A*	0	....	....	....	....	176	203	273
4½	4½ *A*	0	122	153	187	220	255		
0	4½ *C*+4½ *A*	0	96	120	147	175	205	235	

Insulation *A* is natural Sil-O-Cel, for temperatures up to 1600 deg. fahr.
Insulation *B* is Sil-O-Cel C–22, for temperatures up to 2000 deg. fahr.
Insulation *C* is Sil-O-Cel "Super," for temperatures up to 2500 deg. fahr.

* Taken by permission from curves prepared by the Johns-Manville Corp., 1931.

(d) **"Dead air spaces"** were formerly often incorporated in wall structures between the inner and outer layers of brickwork, but these have been definitely eliminated in modern practice. Many tests[29] have shown conclusively that a considerable amount of radiant energy is transmitted across such air spaces and that, in addition, the spaces are usually wide enough to permit the production of convection currents which also transfer some heat. The transmission of radiant energy is of special significance in furnace walls because the sides of the air spaces are usually at elevated temperatures and the quantity of energy radiated is proportional to the difference of the fourth power of the absolute temperatures of these emitting and absorbing surfaces. It is evident, therefore, that a supposedly "dead" air space in a furnace wall may actually increase the rate of heat transmission above that produced in a solid wall of equal thickness. The double-wall construction may be advantageous, however, if the intervening space is filled with ashes or other suitable material, as explained in Sect. **460** (**b**), p. 614.

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[29] Ray and Kreisinger, "Loss of Heat from Furnace Walls," U. S. Bureau of Mines, Bulletin 8, 1912.

(**e**) The **forced cooling of furnace walls,** either by water or air, has as its chief object the reduction or elimination of refractory maintenance. However, the heat absorbed by the cooling medium may be utilized in improving the energy balance of the plant; consequently a water-cooled or air-cooled wall may be looked upon as a device for obtaining a controlled transfer of heat, and in any such wall the transmission of energy to the cooling medium is influenced to a great extent by the circulation of this medium through the wall structure.

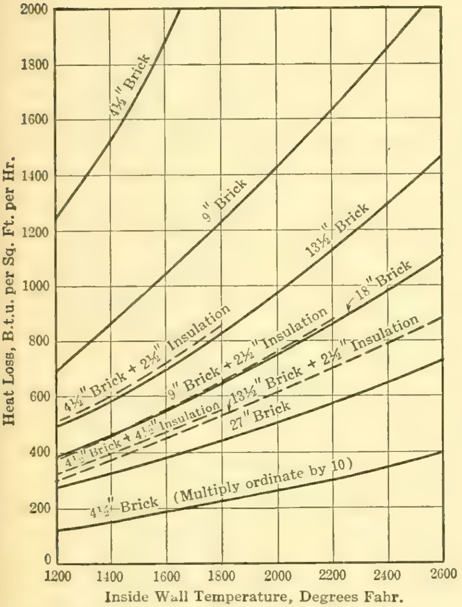

FIG. 702.—Heat Loss through Furnace Walls.[30] (For atmospheric pressure in furnace.)

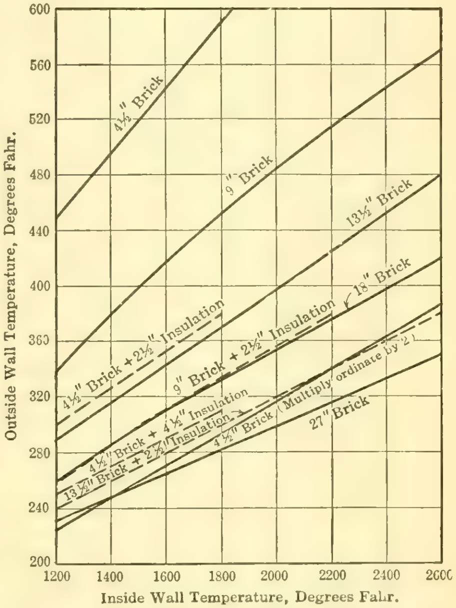

FIG. 703.—Temperature Relations of Faces of Furnace Walls.[30] (For atmospheric pressure in furnace.)

(**f**) The **thermal characteristics of water-cooled walls** must be carefully considered when a choice is to be made between a wall having a bare metallic surface exposed to the furnace gases and one in which the metallic transmitting elements are protected by a refractory facing. In general, the exposed surface of an element of a protected water wall is at a temperature exceeding that of the water within the wall by an amount dependent (1) on the temperature and character of the furnace gases and of the fuel and other solid bodies " seen " by this surface; (2) on the heat-absorbing properties of the surface; (3) on the thicknesses and heat transmission coefficients of the protective coating, joints (or air gaps), tube metal, scale, and surface films; and (4) on the temperature of the inner-tube surface, as maintained by the rate at which the cooling water conveys away the heat transmitted to it. No part of the material should be allowed to attain a temperature that would cause its failure. Thus, when designing a wall, it is desirable to have knowledge of the temperature gradient that will probably be found under the most severe conditions that may occur in service. These gradients through existing furnace walls can be determined by actual measurement; but as yet very few data thus obtained are available. Examples of experimentally determined gradients through some typical bare and refractory-faced Bailey walls, when subjected to the respective rates of heat transmission indicated on the diagrams, are shown in Fig. 704. When the necessary experimentally determined data are not available, the wall temperatures and gradients can be approximated by the analytical methods given in Chapter XXVI, Heat Transmission.

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[30] Reproduced by permission from "Industrial Furnaces," by W. Trinks, John Wiley & Sons.

The *temperature gradient* through the *metallic portions* of water-cooled furnace walls increases progressively as the thickness or refractoriness of the protective facing is reduced, and reaches its highest value in the bare-tube type for which the rate of heat transmission is also a maximum. For example, in Fig. 704 the drop in temperature through the metallic portion of the wall is seen to be about 60 deg. in type WAB; but in type C, which is without protective facing, the drop is about six times as great.

This temperature gradient causes internal stresses in the metal which must be taken into account when the wall is designed. In a high-pressure unit containing bare water walls the temperature of the cooling water is greater, and the wall tubes are thicker (unless of proportionately smaller diameter), than in a low-pressure unit; hence with the former there is greater danger of overheating the outer surface of the tubes, especially if the inner surfaces are covered with films of scale or vapor, or both. The combination of high pressures with bare water walls therefore calls for the use of thin tubes of small diameter, comparatively pure cooling water, and a circulation of the cooling medium rapid enough to prevent films of steam from adhering to the tubes; and further, it is also important to keep the intensity of stress in the outer surface of the tube below the " creep limit " of the material when at the surface temperature, so that the tube will not progressively increase in size and, at the same time lose its desirable physical properties, until it fails after an unduly short period of service. If the operating pressure is 1400 lb. per sq. in., the saturation temperature is 587 deg. fahr., and the outer surface of even a fairly clean tube may be as high as 900 deg. fahr.; and if there are non-conducting films on the inner surface of the tube wall, this temperature may be much greater.[32] Obviously, the rates of heat transmission by water-wall tubes should be limited to values giving safe temperatures in the surface metal.

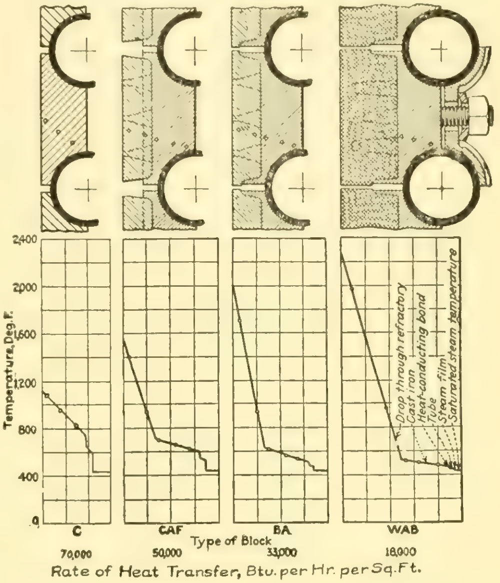

FIG. 704.—Temperature Gradients through Bare and Refractory-Faced Bailey Blocks for Various Rates of Heat Transmission.[31]

(g) **The safe rate of heat absorption by water-wall tubes,** 4-in O.D., 0.165-in. thick (No. 8 gage), has been shown by experience to be 85,000 B.t.u. per hr. per sq. ft. of projected area when exposed to radiation,[33] and, in some examples taken from practice, it was found that, regardless of the tube diameter or steam pressure, the rates used in each case investigated were related to this value about in accordance with the inverse ratio of the wall thicknesses. Pending further investigation, these values may be used as tentative limits of transmission rates for bare tubes made of the usual material. Coutant also states that, compared with the direct radiation absorbed per square foot of the projected area of the exposed boiler tubes, the side walls absorb five-sevenths and the bottom tubes (i.e., screens or floors) absorb one-half as much per unit area. However, the relative distribution of heat to the several walls depends on the method of firing, burner and furnace arrangements, and other factors.[34] The relative values of heat-absorption rates that may be expected with the several different kinds of water walls made by one concern are given in Table LXII. In this table, the assumed standard for reference (taken as 100) is a unit area of furnace envelope that consists of a refractory wall protected on the furnace side by 4-in. tubes spaced with centers 7 in. apart. The ranking given in the last column for the construction there shown, in which the blocks are of silicon carbide having relatively high conductivity, is low because of the poor contact between the loosely fitting blocks and the tubes.

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[31] From "Some Factors in Furnace Design for High Capacity," by E. G. Bailey, Trans. A.S.M.E., 1928, FSP–50–33, p. 266. The following comments on these data are made in Bureau of Mines Bul. 334, "A Study of Refractory Service Conditions in Boiler Furnaces," by Ralph A. Sherman (1931): "The temperatures of the furnace were not given but have been calculated on the assumption of an emissivity of 0.9. It is improbable that the assumption is correct enough to give exact values, but the values are close enough for comparison with previous calculations and data.

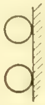
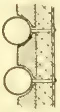
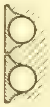
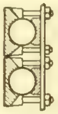
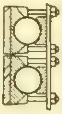
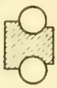
"The type CAF block was stated to be of silicon carbide; the mean thermal conductivity through it was about 90 B.t.u. per square foot per hour per °F. per inch; the surface temperature was 1,550° F., and the calculated furnace temperature was 2,180° F. The material of the BA and WAB blocks was not specified but, as their calculated conductivities were 30 and 26, respectively, it was probably alumina. The temperature of the hot face of the BA block was 2,000° and the calculated furnace temperature was 2,300° F.; the temperatures of the hot face and the furnace for the WAB block were, respectively, 2,300° and 2,450° F.

"It is worthy of note that fire-clay refractories were not used for these blocks. If they had been used, the thickness for the same rate of heat transfer would have been one-ninth that of the CAF block and about one-third that of the BA and WAB blocks, which would have been too thin to be made in this way."

[32] For a discussion of experiences with high-pressure, bare, water-wall tubes see "Operating Experiences with 1300 lb. Steam Pressure," by J. Anderson, Engineering (London), Jan. 6, 1928, p. 25.

TABLE LXII *

RELATIVE RATES OF HEAT ABSORPTION

	Bare tubes	DRAKE c i waterwall blocks	DRAKE Armor-clad water wall	DRAKE Thermo-Fit metal blocks	DRAKE Thermo-Fit refractory faced	Silicon carbide blocks inserted between tubes
Type of wall						
Per cent heat absorption (Bare tubes = 100%)	100	80	70	67½	50	20

* By courtesy of the Drake Non-Clinkering Furnace Block Co.

A **fin-tube wall** probably has not much more heat-absorbing capacity per square foot of projected area of furnace wall than has the assumed standard when clean, because the backs of the tubes in the latter construction receive radiated and reflected heat from the refractory wall behind them nearly equal in amount to that absorbed by the fins. However, the fin tubes have the advantage of giving quite complete protection to the refractory backing, and over a long period of operation are probably more effective than the assumed standard when the space back of the latter becomes filled with ash.

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[33] According to Coutant, as quoted by S. M. Finn, in "Circulation in Water Walls," Engineering and Finance, Nov., 1929, p. 262.

[34] See "Effect of Radiant Heat on Water-Cooled Furnace Walls," by A. G. Christie, Power, May 31, 1927, pp. 841–844.

(**h**) The **thermal characteristics of air-cooled refractory walls** are influenced by a number of factors, the most important of which involve the complex processes of heat transmission in all its forms. Analysis shows that the transfer of heat from the hot surface of the refractory lining to the cooling air is controlled primarily by the *cool surface* of this lining with its accompanying air film. Energy is transmitted from this cool surface both by radiation and convection,[35] the amount of each depending upon the temperature of the surface; and at any section of the air passage this surface will tend to reach an equilibrium temperature such that the quantity of heat transferred to it by conduction through the refractory is exactly equal to that transmitted from it by combined radiation and convection. Except for a comparatively small amount of heat which is transferred through the insulation from the air passage to the room, all the energy transmitted from the cool surface of the refractory is ultimately absorbed by the air flowing through the passage; consequently the temperature and velocity of the cooling air increase progressively, and this results in an increase in the temperature of the cool surface of the lining, between the entrance and exit ends of the air passage. Assuming that steady operating conditions exist in the furnace, any increase in the cool-surface temperature tends to *increase* the rate of heat transmission from the surface and to *decrease* the rate of heat transfer by conduction through the lining; hence the rise in the cool-surface temperature from the entrance to the exit end of the passage produces a different set of equilibrium conditions at each successive passage section. The thermal relations existing in air-cooled walls under various operating conditions have been determined analytically by Kolb,[36] from whose work Figs. 705 and 706 have been taken to illustrate two typical cases. The data for these two figures were obtained on the assumption of perfect thermal insulation between the air passage and the room. Note that in these figures the hot-surface temperature has been taken as 2800 deg. fahr., which is near the upper limit for refractory walls of any type.

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[35] R. H. Heilman, "Surface Heat Transmission," Trans. A.S.M.E., 1929, FSP-51-41, pp. 287-302.

[36] See footnote, page 640.

One of the results of the continuous change in equilibrium conditions between the entrance and exit ends of the air passage is a progressive decrease in the rate of heat transmission from the hot surface of the refractory to the cooling air. The amount of this decrease, per foot of passage length, is related to the other variable factors which have already been mentioned, but has comparatively little significance from an industrial standpoint; consequently the *average* rate of heat transmission for the entire passage length is usually stated. For the example given in Fig. 705, this average rate is about 2600 B.t.u. per hr. per sq. ft. of wall surface, and the corresponding average overall coefficient of heat transfer from the hot surface of the refractory to the cooling air is 1.01 B.t.u. per hr. per sq. ft. per deg. fahr. A table prepared by Kolb shows that the value of this overall coefficient is about unity for all the common examples of air cooling, when a fireclay lining is used. For the example given in Fig. 706, for a wall using a silicon carbide lining, the average rate of heat transmission is about 28,000, and the corresponding average overall coefficient is 6.3.

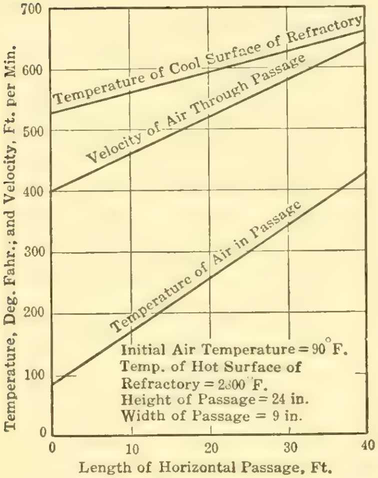

Fig. 705.—Thermal Relations in an Air-Cooled Wall Having Fireclay Refractory 9 in. Thick, and an Initial Air Velocity of 400 Ft. per Min.

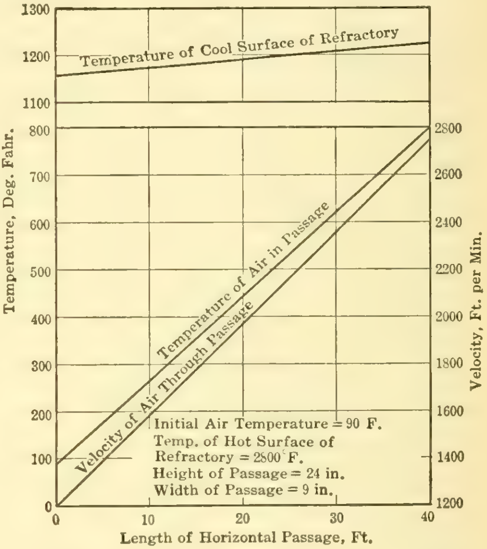

Fig. 706.—Thermal Relations in an Air-Cooled Wall Having Silicon Carbide Refractory, 9 in. Thick, and an Initial Air Velocity of 1200 Ft. per Min.

When an air-cooled wall is to be installed primarily for the purpose of furnishing quantities of air at a definite temperature for combustion, it may be designed to obtain the same final air temperature through the use of widely differing combinations of variables. However, in actual operation, the rates of combustion produced in the furnace cover such a wide range that it is almost impossible to maintain at all times the temperature at the hot surface of the lining for which the wall was designed. Therefore, in practice, the material and thickness of the refractory lining and the arrangement and dimensions of the wall passages are usually determined by structural requirements, and the initial air velocity may be adjusted when the wall is in service in order to produce the required air temperature rise for different rates of heat release in the furnace.

469. Influences of Various Factors on Furnace Design and Operation.—(a) Wohlenberg, with others, has investigated analytically the effect of varying the more important factors that influence the design and operation of furnaces, and has given the methods, data, and results in several papers presented to the A.S.M.E.[37] These investigations are too extensive to permit of detailed consideration in this book; however, some of the more highly instructive charts and conclusions given will be discussed briefly in this section, as showing in a general way the effects of varying the more influential factors.

(b) The **following assumptions** were made for these analyses: (1) the furnace is a cube (usually 20 ft. × 20 ft. × 20 ft.) and is always completely roofed with boiler tubes; (2) the energy release rate (R) and the flame temperature (t_U) are uniform throughout the furnace volume; (3) all the wall surfaces of the same kind exposed within the furnace are at a uniform temperature; (4) the temperature of the gases leaving the furnace and coming into contact with the first convection surfaces of the boiler equals the *mean* flame temperature t_U; (5) the temperature of the water-cooled surfaces is 900 deg. fahr. abs.; and (6), with powdered coal, the flame fills the furnace cavity to within 6 in. of the furnace walls.

Five furnace arrangements (A to E inclusive) were considered, but the results from furnace C are omitted as not contributing to the information afforded by the others. The locations of the water-cooled surfaces in the various arrangements are as follows: Furnace A, top only; Furnace B, top and bottom; Furnace D, top, bottom, and two opposite sides; Furnace E, completely water cooled. The walls that were not water cooled were considered to be of solid refractory and perfectly insulated externally.

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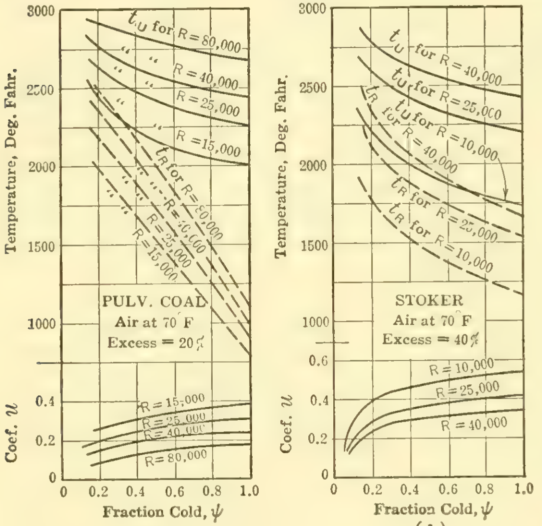
[37] Wohlenberg and Morrow, "Radiation in the Pulverized-Fuel Furnace," Trans. A.S.M.E., 1925, p. 127. W. and Lindseth "The Influence of Radiation in Coal-Fired Furnaces on Boiler Surface Requirements, and a Simplified Method for its Calculation," Trans. A.S.M.E., 1926, p. 849. W. and Brooks, "Some Fundamental Considerations in the Design of Boiler Furnaces," Trans. A.S.M.E., 1927, FSP-50-39, p. 141. W. and Anthony, "Influence of Coal Type on Radiation in Boiler Furnaces," Trans. A.S.M.E., 1929, FSP-51-36, p. 235.

Even though the conditions within an actual furnace may depart considerably from those assumed, nevertheless, the numerical results obtained from these analyses are at least sufficiently exact to lead to valuable general conclusions.

(c) Some **charts**, shown in Figs. 707 to 711, are redrawn from those given by Wohlenberg and Brooks for the case in which the fuel is assumed to be Illinois bituminous coal having a heating value of 12,800 B.t.u. per lb., an ash fusing temperature of 2000 deg. fahr., and certain specified proximate and ultimate analyses.

Fig. 707 (*a*) shows how changes in the fraction cold (ψ), and energy release rate (R) in B.t.u. per hr. per cu. ft., affect (1) the radiation mean temperature (t_U) of the gases in the furnace, (2) the mean temperature (t_R) of the inner surface of the refractory, and (3) the ratio (u) of the energy absorbed by furnace walls to the energy released—all when the coal is burned in *pulverized form* with 20 per cent of excess air supplied at 70 deg. fahr.[38] Fig. 707(*b*) shows the corresponding relation when the furnace is *stoker fired* with 40 per cent excess air at 70 deg. fahr. These curves also apply closely for a furnace volume of 27,000 cu. ft. when using West Virginia coal with a heating value of 14,500 B.t.u. per lb. and an ash fusing point of 2500 deg. fahr., except that the values of u are a little higher than those shown in the figure, for a volume of 8000 cu. ft. Other results not shown here indicate that, if the temperature of the air supplied is raised from 70 deg. fahr. to 500 deg. fahr., the values of t_U and t_R are increased not over 150 deg., and that u is raised about 0.08.



(*a*) (*b*)

FIG. 707.—Influence of ψ and R on t_U, t_R, and u.

Fig. 708 illustrates the influence of energy-release rates and fractions cold on the furnace-flame temperature (t_U) with air supplied respectively at 70 and 500 deg. fahr., and with both *powdered coal* and

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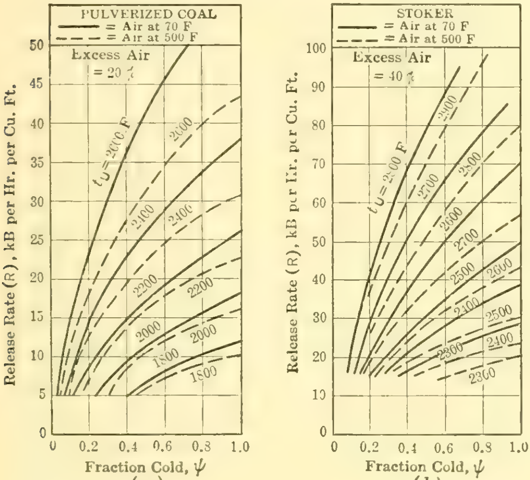
[38] The symbol u is here used for the quantity represented by μ_5 in the original paper.

stoker firing. The curves show how the permissible rate of energy release (R) for a given limiting ash-fusion temperature may be increased as the fraction cold (ψ) is made larger. With a given t_U, the value of R can be more than doubled if ψ is increased from 0.2 to 1.

Fig. 709 shows for *powdered coal* the influence of the percentage of excess air on the placement of the flame-temperature curves with respect to the R–ψ coordinates, with air admitted both at 70 and 500 deg. fahr. respectively. With a given value of t_U, higher release rates or smaller fractions cold can be adopted in the initial design if it is planned to use more excess air; but this is at a sacrifice in the efficiency of the unit.



(*a*) (*b*)

FIG. 708.—Influence of Air Temperature on Furnace-Flame Isotherms.

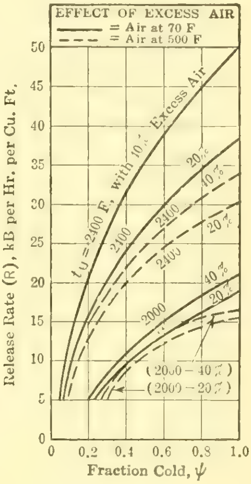


FIG. 709.—Influence of Excess Air.

Also, in operating a given furnace, the release rates or furnace capacity can be increased without exceeding the limiting temperature, if a larger excess of air is used.

In Fig. 710, Charts I and II give some refractory isotherms (t_R) with respect to release rates and fractions cold, for *stoker* and *pulverized coal firing* with 40 and 20 per cent excess air respectively, and with air entering at 70 deg. fahr. The curves shown can be obtained directly from Fig. 707. With air supplied at 500 deg. fahr. the refractory temperatures would be not more than 150 deg. higher than those given by these curves. In Chart III of the figure, the 1800 deg. isotherms for the two methods of firing are superimposed to show more clearly the comparative values of release rates and fractions cold required to maintain this refractory temperature.

Fig. 711 shows the *influence of the furnace volume* on the flame temperature (t_U), with various rates (R) and with air temperatures of 70 and 500 deg. fahr. when the fuel burned is pulverized coal and the furnace is completely water cooled. The effect of furnace volume on the refractory-wall temperature (t_R) is somewhat similar. It is seen that, when pulverized coal is used, the larger the furnace the less does an increase in volume affect the flame temperature. When the furnace is stoker fired, it was found that changes in the magnitude of the furnace volume have a negligible influence on these temperatures.

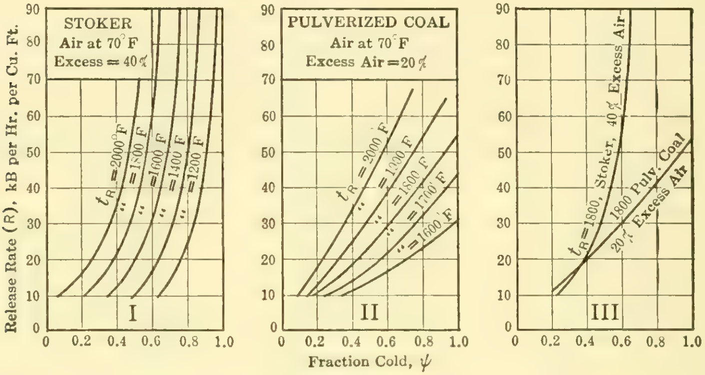


FIG. 710.—Furnace-Refractory Isotherms.

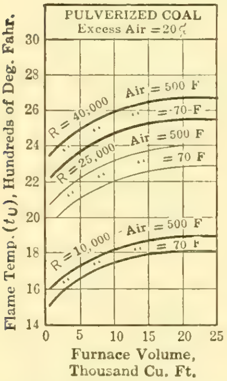


FIG. 711. — Influence of Volume of Furnace, with ψ = 1 and Pulverized Coal.

(**d**) The **influence of the type of coal** on the flame temperature and the transfer of heat within furnaces has been analyzed by Wohlenberg and Anthony,[39] who found that, of the properties investigated, the calorific value of the coal serves as the most significant index of this influence when the coal is burned in powdered form. From the results presented in their report for furnaces having various values of fractions cold (ψ) and energy-release rates (R), and in which different kinds of pulverized coal were assumed to be burned with 20 per cent excess air supplied at 500 deg. fahr., it appears that the following conclusions may be drawn:

(1) When the heat transfer rate (q/A) is around 160,000 B.t.u. per hr. per sq. ft. of projected bare-iron water-cooled surface, it changes about 6000 with each 1000 B.t.u. per lb. alteration in the heating value

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[39] See footnote, page 659.

(F) of the fuel, the energy-release rate being maintained constant; and when q/A is around 20,000, its change with variations in F is negligible.

(2) When the coal has a heating value around 10,000 B.t.u. per lb. and the flame temperature is in the neighborhood of 3000 deg. fahr., this temperature t_U is altered about 50 deg. for each 1000 B.t.u. per lb. change in F, the energy release rate (R) remaining unchanged; and when t_U is about 2000 deg. fahr. the rate of change in t_U with constant R is about 25 deg. with the same variation in F. With coal having a heating value still higher than 10,000 B.t.u. per lb., the rate of change in t_U with variations in F is less, and with poorer coal it is larger, than the values given above.

The changes which occur in q/A and t_U in the intermediate ranges of their values can be approximated by interpolation.

(e) **The effect of adding a refractory coating** to the water-wall surface of a furnace is to improve the combustion conditions when operating under the lighter loadings and, thus, to increase the permissible range of load variation for the furnace—although a somewhat larger furnace volume would be needed to obtain the same maximum capacity as before. Compared with uncooled refractory walls, the refractory-coated water-cooled ones permit the employment of higher flame temperatures and the development of greater capacities per cubic foot of furnace volume. The influence of the refractory coating on water-cooled walls has been investigated analytically by Wohlenberg and Brooks, and the results were reported in their paper to which reference has already been made. Some of these results will now be considered briefly for walls which have flat, water-cooled metallic surfaces to which the refractory coating is attached with intimate contact. It is found that water cooling has little thermal advantage when the coating is more than 2 in. thick and of the usual refractory material, i.e., one having a coefficient of conduction of about 10 B.t.u. per hr. per deg. fahr. per sq. ft. per in. The effect on t_U, t_R, and u of varying the fraction cold (ψ) and the thickness of the refractory coating of 2 in. or less is shown in Fig. 712 for the conditions stated in connection therewith; and the effects on t_U and t_R of varying the energy release rates and the refractory thickness are shown in Fig. 713, for a furnace which is completely lined with walls of this type, except at the top face, which consists of the bare boiler tubes. Changing the temperature of the entering air from 70 to 500 deg. fahr. was found to increase t_U about 50 deg.; and to raise t_R about 25 deg. when the refractory thickness is $\frac{1}{2}$ in., and about 100 deg. when the thickness is 2 in. A thickening of the refractory coating requires the use of a larger furnace volume if the same flame temperature is to be maintained. For example, to obtain a certain temperature in a furnace, a release rate of 30,000 B.t.u. per hr. per cu. ft. would be required with bare water walls, 23,000 with walls having a half-inch refractory coating, and 16,000 when the coating is 2 in. thick. The relative furnace volume required in the three cases would, of course, be inversely proportional to the release rates. Thinning down the refractory coating has the effect of narrowing the range of economical load variation, but permits a reduction in the size of furnace volume required.

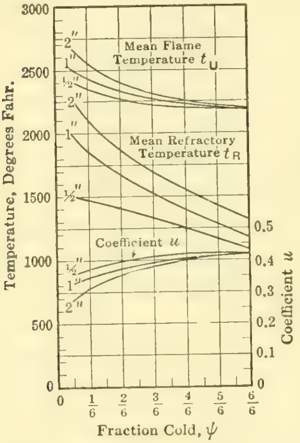

FIG. 712.—Effect of Varying the Fraction Cold and Refractory Thickness when R = 25,000 B.t.u. per Hr. per Cubic Foot.[40]

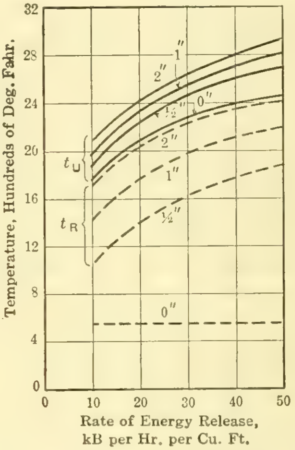

FIG. 713.—Effect of Varying the Refractory Thickness and Energy Release Rate in a Furnace[40] Having $\psi = \frac{1}{6}$.

(f) The following **fundamental principle in furnace design** was stated by Mr. E. G. Bailey in discussing the paper of Wohlenberg and Brooks: "*The best furnace is one, the conductivity of which is such that all walls will be at a temperature equal to the fusing temperature of the ash.* Cooler walls have a detrimental effect on combustion, while wall surfaces hotter than this cannot be maintained, as the washing of the slag will automatically bring the refractory thickness to such a point that its surface temperature will equal that of the fusing temperature of the ash."

He also pointed out that this paper shows only the mean temperatures that may be expected, whereas, actually, hot spots that may be at temperatures 300 to 400 deg. higher than the mean are experienced, and more cooling and more fraction cold must be provided at these locations if adequate protection is to be afforded.

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[40] For the following conditions: Furnace, 20 ft. × 20 ft. × 20 ft.; air, 70 deg. fahr., 20 per cent excess; coal, Illinois or West Virginia; conductivity of refractory, k = 10 B.t.u. per hr. per sq. ft. per in. per deg. fahr.; for any other conductivity, multiply the refractory thickness by $0.1k$.

CHAPTER XXXIX

BOILERS

470. General.—(**a**) The term **boiler** was formerly used to cover the aggregation of the boiler proper and the other closely related parts that function with it in the generation of steam, and later the term " boiler unit " was also applied to this aggregation, especially when supplementary heat-absorbing apparatus was included. It is now the preferred practice to restrict the application of the term **boiler** to the more or less distinct heat-absorbing element which has for its sole purpose the conversion of the working substance within it from a liquid state into a vaporous one, and to use the term **steam-generating unit** [1] for the larger aggregation.

In the more usual forms of steam-generating units, the various kinds of heat-transmitting elements are quite distinct from each other, and frequently are constructed separately, often by different manufacturers, and are not connected together until the unit is assembled at its destination. In units of this kind the boiler is a clearly defined element, and it is to such apparatus that the first part of this chapter is primarily devoted. In some of the more unusual arrangements of steam-generating units, on the other hand, there are no clearly defined points of demarkation between the various portions of the apparatus that function respectively as economizer, furnace water walls, boiler, and superheater. Some forms of apparatus of this kind, which must be considered mainly as a unit, will be discussed later in the chapter.

(**b**) Briefly, the usual **boiler structure** is composed mainly of iron or steel tubes or flues, and one or more cylindrical shells or drums, these elements being joined together either directly or through headers. The tubes, commonly of from 2 to 4 in. outside diameter (O.D.), are connected to the other parts by inserting their ends into the tube holes bored in those parts and " expanding " the ends sufficiently to form very tight fits therein. The drums or shells are usually constructed of steel plates, which in the past were always connected by means of riveted joints, but now welded seams may be substituted, and for use with very high pressures the drums are solid-forged. Of the elements that can be used in constructing pressure vessels, those of cylindrical form have the greatest inherent strength (except spherical ones, which, however, are usually of unsuitable shape or are difficult to employ in boilers), and flat plates are the weakest; hence, when possible, the former are adopted in preference to the latter. Plates that are flat, or nearly so, and have large areas subjected to high pressure must be either of great thickness, or, if thin, must be stayed by some means, as by using stay bolts, stay rods, gusset (plate) stays, or reinforcement girders. Ordinarily, tubes with the expanded joints have adequate holding power to act as stays for the portions of plates to which they are attached; but, in special cases, stay tubes having other more effective methods of attachment are substituted. As the rupture of small elements causes less damage than that of large ones in a boiler explosion, preference is naturally given to using the former when otherwise feasible. In addition to its main structure, the boiler may be considered also to include the safety valve, the steam gage, the water gages, and the other accessories that are directly attached to it, and likewise the connections and valves that are provided for supplying the feedwater, for delivering the steam, and for " blowing down " to remove scale deposits or partly or wholly to empty the boiler.

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[1] See page 430 for the definition of a steam-generating unit.

471. Boiler Requirements.—(**a**) The more **important requirements** which are necessary or desirable for boilers to meet will now be considered. Naturally the variety of conditions under which boilers are operated and the importance of continuity of service in different plants affect somewhat the order in which these requirements would be rated in any specific case.

(**b**) The **general suitability** of a boiler is determined by its *design*, *size*, and *proportions*, which should be such as to permit the element to be forced to the degree necessary to meet not only the usual maximum load, but also emergency demands. The *arrangement* should be suitable for the employment of the boiler and furnace with the kind or kinds of fuel to be burned, the fly-ash to be encountered, and the character of the feedwater and of the draft available; and the *form* of the boiler should be such as to make it adaptable to the kinds of grates, stoker, or other type of burner used, to the desired size and proportion of the furnace, and to its combination with such superheater, economizer, reheater, and water-wall elements as are to be included in the unit.

(**c**) The **safety and durability** of a boiler depend fundamentally on the design of the apparatus and the materials and workmanship employed in its construction. Not only must the parts be able to withstand with ample margin of safety the normal steam pressure and temperature to which they are subjected, but the arrangement of elements and the method of support, or of suspension, of the structure as a whole should be such as to avoid any possibility of having serious stresses induced in the parts by inequalities in expansion or contraction that may accompany temperature variations. In some designs the use of curved tubes gives a desirable degree of flexibility to the structure and provides adequately for the expansion differences. There should be no thick plates or other thick parts (such as lapped joints), and no projecting portions or plate edges, exposed to the impact of the flame or intense radiation; nor should the blowoff pipe be similarly exposed.

The materials, design, and construction of the boiler, and also of the other pressure elements of a unit, should be at least as good as the standards of practice specified in the " Boiler Construction Code " and in the " Interpretations of the Boiler Code " which the A.S.M.E. has prepared and frequently revises, and which have been adopted as legal requirements by most of the political divisions of our country; also there should be conformity to any local ordinances and to insurance regulations to which the boilers may be subjected.

(d) The **accessibility** of all parts of a boiler is important for inspection, cleaning, or making repairs. Suitable doors should be provided in the boiler setting for reaching all exterior parts of the heating surfaces, and the boiler structure must have suitably located manholes and handholes to give the necessary access to all interior parts.

In comparing boilers as to provisions made for *internal cleaning*, cognizance must be taken not only of the number of manhole and handhole joints to be broken and subsequently made tight, but also of the time required for doing this, for cooling the boiler and its setting sufficiently to permit of such work, and for bringing the boiler into operation again. The *external cleaning* of water tubes, and the removal of loose dust and scale from the inside of fire tubes, may be accomplished by means of blasts of steam or compressed air issuing from small hand-directed tubes or " lances." To provide for inserting the lances, openings are placed in the fronts and backs of some boilers or their settings; in other arrangements, the openings are placed in the side walls of the setting. Sometimes permanently installed " soot blowers " are included as part of the boiler equipment.

The design of the boiler should be such as to permit repairs to be made easily. In most types of boilers the principal structural difficulties develop in connection with the tubes. Thus, the arrangement should be such as to permit the ready removal of any one of the tubes without disturbing the other tubes or other parts. If the tubes are straight, only a few spare ones need to be carried in stock; but if they vary widely in curvature, a large assorted collection may be needed to meet any emergency that may arise.

(e) The **water circulation** in boilers of the usual types depends on natural circulation of the water within them, and to provide for it there should be definite and unrestricted circuits for the liquid to follow, as in Fig. 714, and the heat should be applied at such places in the apparatus as will establish and maintain an adequate rate of flow under all conditions of operation. The arrangement of boiler elements should be such that there can be no stagnant water or steam in contact with highly heated surfaces and that no detrimental short-circuiting can occur.

(f) The **gas passages** through a boiler should be such as to maintain suitable velocity of the gases, which should proceed through the boiler without sudden contraction or expansion. Within limits, the greater the velocity the more rapidly will the heat be conveyed to the heating surface and the greater will be the amount of evaporation from the surface swept over. There should be no pockets where the gas can remain stagnant, and it is desirable to have the gas baffled in such a way as to bring constantly the fresher portions into contact with the heating surface as the gas proceeds.

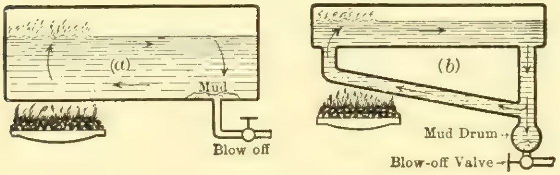

Fig. 714.

(g) **Dry steam** is always desired from a boiler. To prevent **priming,** i.e., the entrainment of a considerable portion of moisture by the steam, the *liberating surface* of the water from which steam is released should be of ample area. When the water contains certain impurities **foaming** may occur, and this always increases the amount of the moisture entrained. By providing a large steam space the life of the particles of steam within the boiler may be made sufficiently long to allow nearly complete precipitation of the moisture. Provision is often made within the boiler for the separation of moisture by means of "dry pipes," baffles, or other steam-separating devices.

(h) The **quantity of water** in a boiler affects several items. If the boiler contains a large volume of water less attention is required to maintain the water level, and the boiler has a greater reserve to meet sudden demands than is the case in boilers having a small volume; but greater damage would ordinarily result in case of explosion. In marine and similar service the greater weight involved is of course objectionable.

(i) The **introduction of feedwater** into a boiler should be in such a manner as not to retard the circulation of the water therein, and, if cold, this water should not be allowed to come in contact with the boiler shell. Certain of the impurities which may be in solution in the entering water precipitate when the higher temperatures are reached and are deposited as mud. The water should be introduced at such a point and the arrangement of boiler should be such that this precipitate will be deposited where it will do the least damage and from which it can be readily removed. See Fig. 714 (*a*) and (*b*). Sometimes a " mud drum " is provided as in the latter figure, or the water is first introduced into a " settling chamber " as in Figs. 730 and 742, from which the mud may be blown off from time to time.

(j) The **space occupied** by a steam-generating unit is an important part of the first cost of a plant. In addition to the floor space and height occupied by the boiler and furnace, there must be charged against the apparatus the amount of space that must be provided for the replacement and cleaning of the boiler tubes. In some horizontal boilers there must be space in front (or rear) at least equal to the length of the tubes to be withdrawn or inserted; hence this may fix the minimum distance between parallel rows of boilers or between the boiler end and the wall of the building. Vertical boilers sometimes occupy less floor space than do horizontal ones of the same steam-generating capacity but may require room above them for tube replacement. If doors, windows (or skylights), or other openings of sufficient size are located in the walls (or roof) of the building in such position that the tubes can be passed through them when retubing is necessary, no additional space need be provided within the building for this operation.

When the exterior of the boiler heating surface is accessible for cleaning from the front (or rear) of the setting the units can be arranged in a **continuous battery**, with adjacent walls in common, thereby saving the space that would otherwise be allowed between independent settings. When the cleaning must be done from the side of the boiler, the units may be independent settings with working space on both sides, or they may be arranged more compactly in **batteries of two units each** and with sufficient space between the pairs to permit of access to openings in one side of each setting.

(k) **The cost** is always an item of fundamental importance. Besides the first cost of the boiler, with its setting and trimmings, and the expense of its transportation and erection, it is necessary to consider also the charges for up-keep and repairs, and the cost of the ground and building space required for the boiler and furnace, including that needed for maintenance.

In connection with the brief study of the various types of boilers which follows after the next section, one should note the various ways in which the different arrangements meet these requirements.

472. The Classification of Boilers.—(a) **Commercial designs of boilers** assume many different forms, which have been developed to a high degree of reliability as the result of long experience with them under widely varying conditions; and these are now considered to be more or less *standardized types*. Most types of boilers, when similarly proportioned, give substantially the same performance under like circumstances, any comparative inequalities in the overall results obtained from steam-generating units being due mainly to differences in operating conditions or in the functioning of parts of the unit other than the boiler.

(b) **The classification of boilers** can be neither simple nor complete, since the factors which distinguish one boiler from another include not only the structural details of each, but also the relation which each bears to the steam-generating unit in which it is installed, as well as to the plant as a whole. However, all boilers fall into certain broad general classes, as outlined below, any one design being included in several of the groups; thus a locomotive boiler might be placed in classes A3, B2(*b*), C2, E1, and F1, being of the internally fired, through tube, horizontal type.

(c) The use of the **working steam pressure** as a basis for classifying boilers is unsatisfactory in most cases, since a pressure which would be considered high in one type of service might be looked upon as medium or low in another; consequently the terms " high pressure," " medium pressure," and " low pressure " may be used only for purposes of comparison within any particular field of service. The absolute value of the working steam pressure is determined largely by the requirements of the equipment which uses the steam generated, and the design of the boiler itself is influenced by this pressure only to the extent that the parts must have adequate strength to meet the conditions imposed.

473. Horizontal Return-Tubular Boilers.—(a) Because of their relatively *low cost*, *compactness*, and *large water-content*, which gives them stable steam-generating qualities, this simple type of boiler is perhaps the most widely used of any in this country in sizes ranging from about 150 to 3750 sq. ft. of heating surface, the corresponding shell sizes varying from 36 in. diam. by 8 ft. long to 90 in. diam. by 20 ft. long. The steam pressures used with them seldom exceed 150 lb. per sq. in. gage, because if the shell plates that are exposed to the furnace are made too thick they will overheat and burn out; and the load intensity is limited to about 5000 B.t.u. per hr. per sq. ft. of heating surface. Boilers of this type have quite simple settings, and any number of these boilers can be arranged in a continuous battery. Settings for these boilers were discussed in Sect. **460**.

CLASSIFICATION OF BOILERS
according to:

A.—Purpose.

1. Heating.
2. Process.
3. Power plant.

B.—Kind of Plant.

1. Stationary, or land.
   - (*a*) Residential.
   - (*b*) Industrial.
   - (*c*) Central-station.
   - (*d*) Portable or Semi-portable.
2. Mobile.
   - (*a*) Marine.
   - (*b*) Locomotive.
   - (*c*) Automobile.
   - (*d*) Airplane.

C.—Method of Firing.

1. By hand.
2. By mechanical stoker.
3. In suspension.
   - (*a*) Powdered coal.
   - (*b*) Oil.
   - (*c*) Gas.

D.—Special Forms.

1. Waste heat.
2. Electrically heated.
3. Submerged combustion.

E.—Working Substance.

1. Water.
2. Mercury.
3. Diphenyl and diphenyl oxide.

F.—Arrangement and Structural Features.

1. Fire-tube.
   - (*a*)
     - Externally fired (Figs. 715 and 716).
     - Internally fired (Figs. 717 to 724 inclusive).
   - (*b*)
     - Through tube (Figs. 717 to 721 inclusive.)
     - Return tubular (Figs. 715 and 716).
   - (*c*)
     - Vertical (Figs. 717 to 719 inclusive).
     - Horizontal (Figs. 715 and 716, 720 to 724 inclusive.)
2. Water-tube.
   - (*a*)
     - Straight tube (Figs. 725, 738, and 739).
     - Bent (curved) tube (Figs. 741 to 745 inclusive.)
   - (*b*)
     - Horizontal tube (Figs. 725 to 737 inclusive).
     - Vertical tube (Figs. 738 and 741A).
     - Semi-vertical (inclined) tube (Figs. 739, 741 to 743 inclusive.)
   - (*c*)
     - Single pass (Figs. 734 to 737 inclusive, 739 and 740).
     - Two-pass (Fig. 738).
     - Three-pass (Figs. 725, 732, and 741).
   - (*d*)
     - Longitudinal drum (Figs. 725 and 730).
     - Cross drum (Figs. 732 to 737 inclusive).
   - (*e*)
     - Single drum (Figs. 732 to 737 inclusive).
     - Two-drum (Figs. 738 and 741A).
     - Three-drum (Figs. 741B and D).
     - Four-drum (Figs. 742 and 743).
   - (*f*)
     - Sectional header (Figs. 725 to 729 inclusive).
     - Box header (Figs. 730 and 731).
   - (*g*)
     - Cross baffled (Figs. 732 and 743A).
     - Parallel baffled (Figs. 730, 738, and 741).
   - (*h*)
     - Vertical mud-drum (Figs. 738 and 739).
     - Horizontal mud-drum (Figs. 740 to 745 inclusive).

(b) The **Structural details** of a typical arrangement of a H.R.T. boiler are shown in Fig. 715. The shells of such boilers are, according to size, made of one, two, or three circumferential rings, with riveting so arranged that the longitudinal seams are not exposed to the fire and that the plate edge of the first circumferential seam does not receive the impact of the flame or heated gases. The boiler shown in Fig. 715 has manholes located above and below the tubes, and its shell is extended to form a projecting smoke box, thus the furnace front (usually of cast iron) covers only the portion of the front wall below the shell and is therefore termed a *half front.* This boiler is freely suspended from *gallows frames,* consisting of cross beams supported by columns which are located outside of the side walls. In addition to the main details of the structure, many of which are labeled, the figure also shows the steam gage, water gage, try cocks, safety valve, fusible plug, feed pipe and valves, dry pipe at the steam outlet, stop valve and non-return valve, and the blowoff valves and pipe.[2] The latter is protected from flame impact and is connected to the low point in the shell, which is set about 1 in. lower at the rear than at the front, the point of attachment also being in the region where most of the mud is deposited in the shell.

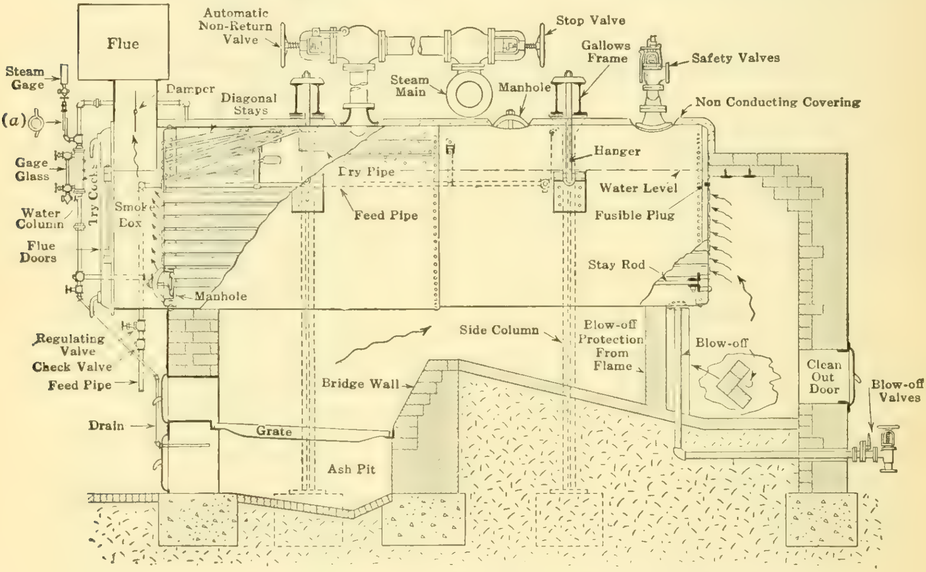

FIG. 715.—Horizontal Return-Tubular Boiler, with Half Front Setting.

Fig. 537, page 431, shows a H.R.T. boiler which is supported by brackets which rest on the side walls of the setting, rollers being interposed between the rear brackets and their bearing plates to provide for the free expansion of the shell. With this boiler the smoke chamber is formed in the brickwork of the front wall, which is covered by a *full flush front.*

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[2] These boiler accessories are discussed in Sect. **484.**

(c) A **modified return-tubular boiler** of the semi-portable type is illustrated in Fig. 716; this arrangement forms a compact, self-contained unit suitable for moderate capacities and for pressures up to 125 lb. per sq. in.

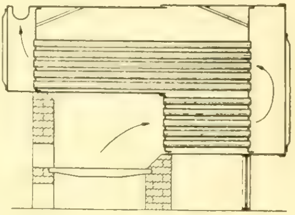

FIG. 716.—Semi-Portable Boiler of the Modified Return-Tubular Type.

474. Internally Fired, Tubular Boilers.—(a) Boilers of this kind are *compact* and *self-contained,* and they are shipped from the factory as completed structures. They cost more than externally fired fire-tube boilers, but they avoid the expense for special brick settings and their upkeep. They are limited to sizes that can readily be transported.

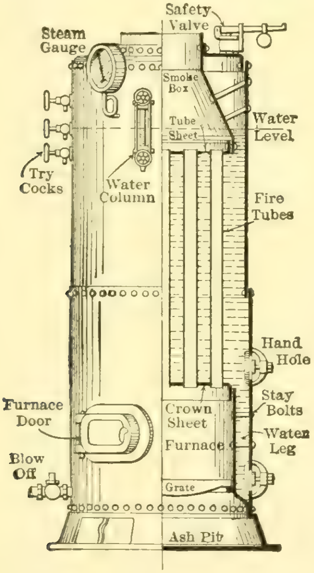

FIG. 717.—Tubular Boiler. Submerged-Tube Type.

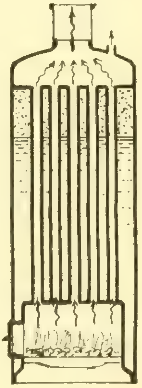

FIG. 718. Tubular Boiler Exposed-Tube Type.

(b) An example of a **small vertical tubular boiler** of this kind is given in Fig. 717, in which the different parts and fittings are labeled. This boiler is of the **submerged-tube type,** for the water level is above the tubes. When the tubes extend into the steam space, as in Fig. 718, the boiler is of the **exposed-tube type.**

The boilers of the kinds illustrated in these two figures are built usually in sizes from 21 to 66 in. in diameter, with 40 to 1000 sq. ft. of surface, and for pressures not exceeding 150 lb. per sq. in. gage. They are somewhat difficult to inspect and clean, have little steam space and limited furnace volume, and the steam pressure may fluctuate considerably with varying demands. In Fig. 717 the principal parts of the boiler proper and its fittings (furnace door, grates, and manholes) are labeled and also the necessary accessories are shown.

(c) **Large vertical tubular boilers** are usually of the exposed-tube type and have different diameters for the upper and lower parts of the shell, as in Fig. 719. They are built in sizes ranging from 500 to 6000 sq. ft. and for pressures up to 200 lb. per sq. in. gage, and deliver steam that is slightly superheated. In the larger sizes the annular space shown at (a) can be occupied by a man when cleaning the tubes, crown sheet, and plates around the furnace. In the *Manning* boiler the upper and lower parts of the shell are connected by means of a *reverse flange* or *Ogee ring*, as shown at (B) in Fig. 719, in order to give sufficient flexibility in the shell to provide for differences in the expansions of the tubes and the outer envelope. Numerous handholes give access to the crown sheet.

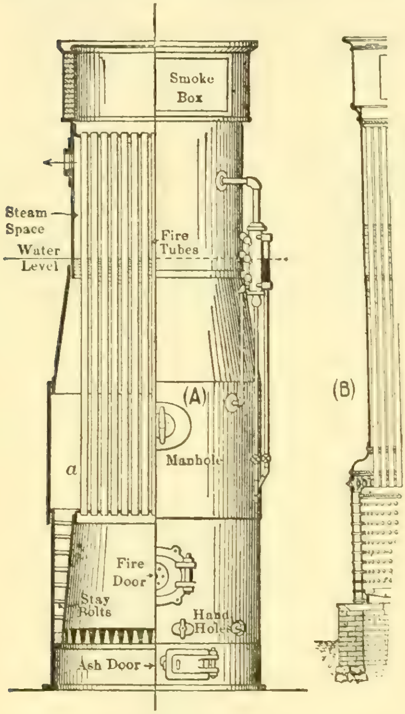

Fig. 719.—Tubular Boiler. Exposed-Tube Type.

(d) **The locomotive boiler** is of the internally fired, fire-tube type, as shown in Fig. 720, and is designed primarily to produce large quantities of steam within restricted space limits. The basic form of this steam-generating unit has become well established over a long period of time and consists essentially of (1) a steel-plate *shell* containing a steel-plate *firebox*, and through tubes, or *flues*; and (2) a cast-iron or cast-steel *smokebox* containing the *draft-producing equipment*. In modern practice the smokebox also houses the *superheater headers* and, in many recent cases, the *throttle valve* and *feedwater heater* as well. In the conventional design, which may be used with steam pressures up to about 300 lb. per sq. in., *stay-bolts* and *stay-rods* are used to support the flat surfaces of the firebox and the back head of the shell against the steam pressure which tends to deform them; however, for pressures above 300 lb. per sq. in., the thickness of the plate must be increased to the point where the use of stay-bolts is no longer feasible, and all flat stayed surfaces are then eliminated in favor of some form of water-tube firebox.[3] Recently nickel steel plates have been substituted for plain carbon steel plates in boilers designed for steam pressures between 200 and 300 lb. per sq. in. This has resulted in the use of thinner plates to withstand a given pressure, and has enabled the designer to retain the stayed firebox for steam pressures around 250 lb. per sq. in. where the water-tube form would otherwise have been necessary. Nickel steel plates may also be used to save weight in the design

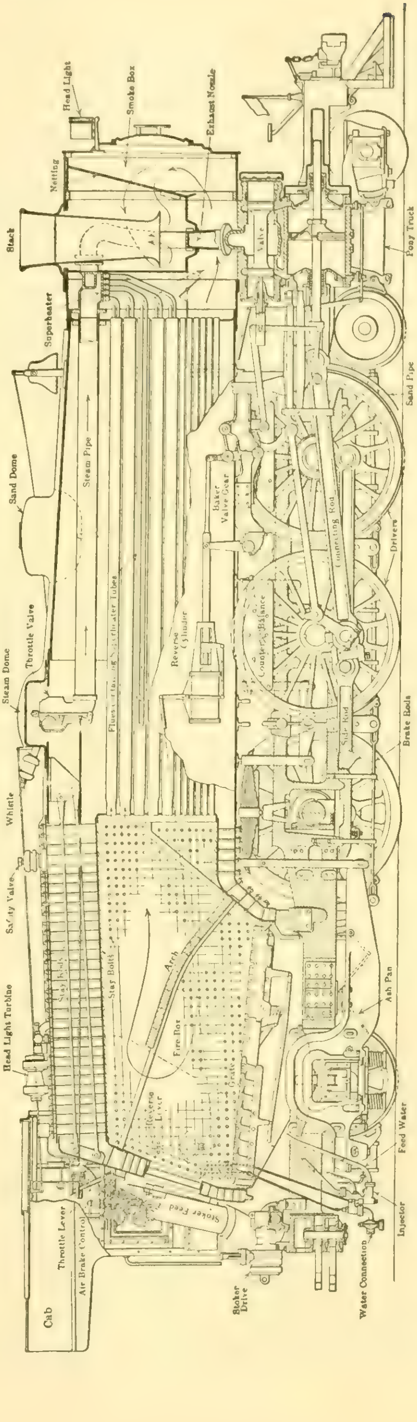

Fig. 720. Locomotive Power Plant—Pacific Type of Passenger Locomotive. (Through the cooperation of R'y and Loco. Eng., N. Y.)

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[3] Excellent descriptions of modern water-tube firebox installations are given in Railway Mechanical Engineer, February, 1925, p. 79, and August, 1931, pp. 397–403, and in "The Locomotive Cyclopedia," Simmons-Boardman Publishing Co. (1927), pp. 251–258.

of the large fireboxes required by locomotives which are intended for high sustained output.

When the water level in a modern locomotive boiler of large size is brought to its normal height, which is several inches above the top of the firebox, the body of water contained within the unit may easily amount to 5000 gal. or more. During operation, a rapid and definite circulation must be established within this large body of water in order that high rates of steam generation may be maintained at the surfaces forming the sides and top of the firebox and combustion chamber, especially when the locomotive is working at capacity for long periods of time. When a water-tube firebox is used, rapid circulation in this part of the unit is an inherent feature of the design; but in the conventional type of firebox, the required result can be obtained only through the use of circulating devices such as arch tubes (shown in Fig. 720) or "thermic syphons."[4] Since these devices extend diagonally upward and backward across the combustion space, rapid circulation is set up within them, by means of which water is drawn from points below the front end of the firebox and discharged at points close to the top of the combustion chamber. A considerable amount of steam is generated within the arch tubes or syphons and is swept upward to the water surface, from which it can easily enter the steam space. The maintenance of a convection current within the body of water as a whole is assisted by the introduction of the feedwater at a point near the front flue sheet, from which it can progress downward and backward towards the firebox.

Most modern coal-burning locomotives are fired by mechanical stokers of the overfeed type and the fuel is burned on shaking grates. The installed grate surface in large motive-power units of this kind varies from 75 to 90 sq. ft., while average combustion rates range from 100 to 150 lb. of coal per hr. per sq. ft. of grate surface, when the locomotives are delivering high outputs. These high rates of combustion are made possible by the intense induced draft, varying from 5 to 10 in. of water, which is produced in the smokebox by the ejector action of the jet of exhaust steam from the cylinders. Since the firebox and combustion chamber are entirely water-cooled, extremely high rates of heat release are possible, an average value for large locomotives under high-output conditions being about 200,000 B.t.u. per hr. per cu. ft. of volume in the firebox and combustion chamber combined.

The comparatively low efficiency of the locomotive steam-generating unit over its entire range of operation is due primarily to incomplete combustion, high exit-gas temperatures, and considerable losses of unburned carbon through the stack. These conditions are caused by the following factors: (1) operation at very high rates of heat release when the locomotive is working at capacity, (2) the use of an intense draft, and (3) the presence of large expanses of water-cooled metallic surface in contact with the burning gases. Although the conditions mentioned above are undesirable for several reasons, the factors which cause them are essential features of the compact, effective, and relatively inexpensive steam locomotive, which is capable of rendering transportation service at the lowest total unit cost over a wide range of operating conditions; hence this type of locomotive should be judged on the basis of its superior overall performance, including its relatively low fixed charges, rather than on the comparatively poor efficiency of its steam-generating unit.

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[4] For a description of the Nicholson Thermic Syphon, see "The Locomotive Cyclopedia," Simmons-Boardman Publishing Co. (1927), pp. 271–272.

Tests of a typical large modern locomotive of conventional design, generating steam at a gage pressure of 250 lb. per sq. in., with a superheat of 250 deg. fahr., from feedwater at 210 deg. fahr., and using coal having a heating value of 12,600 B.t.u. per lb., as fired, would show an evaporation of about 6.5 lb. of water per lb. of coal, under high-output conditions involving the production of 70,000 lb. of steam per hr. The efficiency of the steam-generating unit would be just 60 per cent in this case.

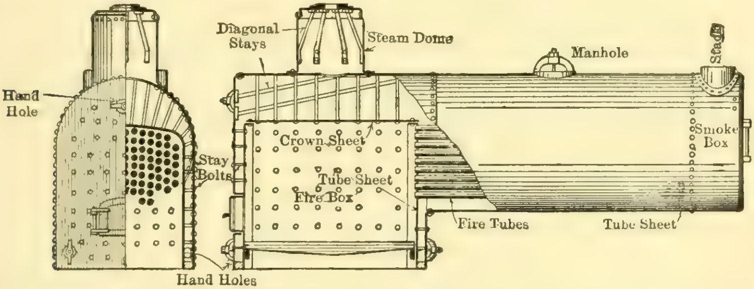

FIG. 721.—Locomotive Type of Stationary Boiler.

The most positive step taken to improve combustion conditions has been the general adoption of the refractory combustion arch for installation within the firebox. This arch is composed of suitably shaped fireclay tiles, supported at their ends on the arch tubes or syphons, and forming a continuous baffle wall over which the products of combustion must pass in order to get to the flues, as shown in Fig. 720. This arch has the effect (1) of increasing the furnace temperature, (2) of preventing the short-circuiting of the gases from the fuel bed to the flues, and (3) of promoting the turbulence and consequent mixing of the streams of air and gases which rise from the top of the fuel bed.

(e) **A stationary locomotive boiler** of the portable or semi-portable type is depicted in Fig. 721, in which the important parts are labeled. The dome provides additional steam space but is often omitted. Boilers of this type are commonly built in sizes ranging from 150 to 2500 sq. ft. of heating surface and for pressures from 50 to 150 lb. per sq. in. gage.

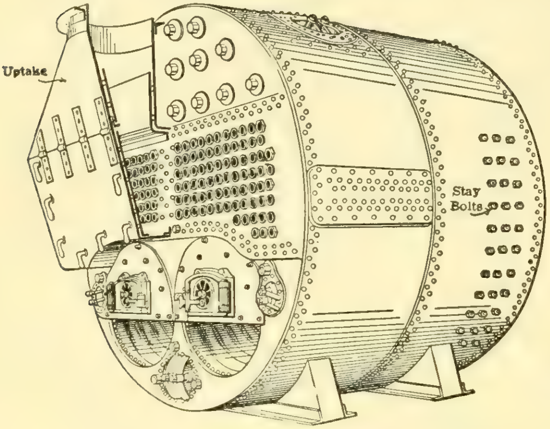

Fig. 722.—Scotch Marine Type of Boiler.

(f) **The Scotch Marine Boiler** is shown in Figs. 722 and 723. The furnace and combustion chamber are part of the boiler, which constitutes a completely self-contained unit. The furnace walls are cylindrical flues which have strength-giving corrugations, and the combustion chamber has metallic walls and is entirely surrounded by water. As the design requires extensive use of flat and convex plates under pressure it is necessary to use many stay-rods, stay-bolts and girders to strengthen these walls. Such boilers have from one to four furnace flues from 7 to 8 ft. long, and their outer shells range from 6 to 16 ft. in diameter. They have been built for pressures up to 300 lb. per sq. in. They contain large quantities of

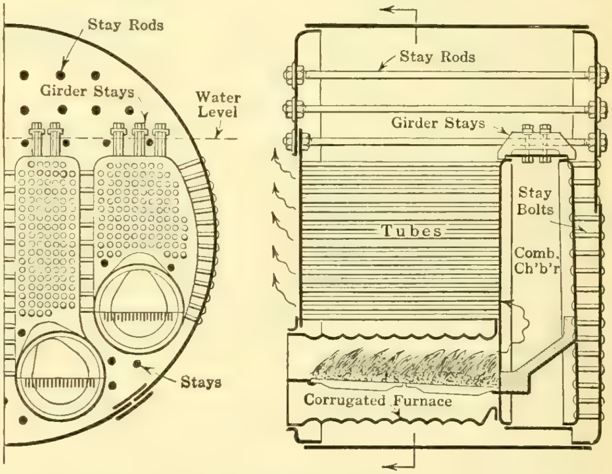

Fig. 723.—Sections through a Single-Ended Scotch Marine Boiler.

water and have voluminous steam space and extensive steam-liberating surfaces. Because they have short tubes, large steaming capacity for the space occupied, and accessibility, and are complete in themselves, they are particularly adapted to marine service and other applications where space is very valuable. Their relatively high cost is offset by the absence of expense for a separate setting.

(g) The **dry-back Scotch boiler,** Fig. 724, is an adaptation of the Scotch marine type to stationary service by substituting a refractory-lined combustion chamber for the submerged one, a change which greatly simplifies the construction and reduces the cost. They are usually built for pressures from 100 to 200 lb. per sq. in. gage, have either one or two furnace flues, and range in size from 60 to 3000 sq. ft. of heating surface, i.e., from 3 to 10 ft. in diameter, with length about double the diameter. They are especially applicable where low head room is essential, as in office buildings and hotels. They are not suitable for use with high volatile coals, and the circulation in them may be defective because the water in the bottom of the shell is rather stagnant and, as a result, the furnace walls and tubes may expand or contract unequally and thus cause tube leakage.

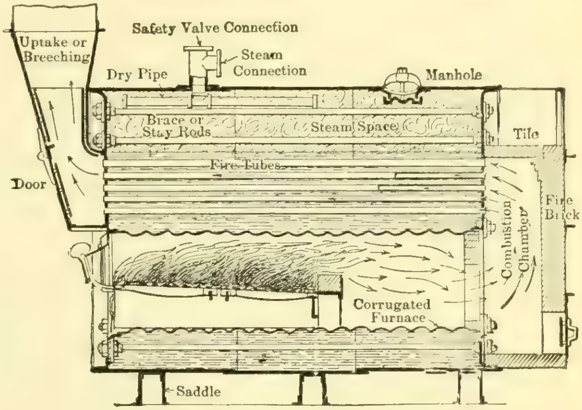

FIG. 724.—Dry-back Scotch Boiler.

475. Longitudinal-Drum Water-Tube Boilers.—(a) Boilers of this horizontal type are constructed primarily of straight tubes, drums and headers, some having *sectional headers* and others *box headers*. This type is one of the most extensively used in this country.

(b) The **sectional-header type,** which was formerly known as the **B. & W. type** of boiler,[5] is exemplified in about its simplest form in Fig. 725; and it has the water circulation and arrangement of mud drum and blowoff shown in Fig. 714(*b*). The boiler is freely suspended from overhead cross beams by means of U-rods, and in constructing it, first the tubes are expanded into front and rear headers to form "tube sections," then these sections are connected to one or more suspended drums by means of nipples or short lengths of tubing which are expanded into counterbored holes in the tops of the headers and bottoms of the cross boxes of the drums. As the sections, drums, nipples, and other parts are shipped knocked-down and are assembled at the power house, the parts transported and handled are relatively small. By varying the number of drums, lengths of tubes, number of sections wide, and number of tubes high, such boilers can be readily adjusted to give the desired capacity and at the same time be of proportions suitable for the furnace arrangement desired. The steam drums, which range in diameter from 36 to 60 in., are constructed without circumferential joints, except those at the ends, and have wrought-steel heads which are dished to a radius equal to the diameter of the drum and require no supplementary

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[5] "B. & W." indicates the Babcock & Wilcox Co., which originated this type of boiler. Formerly each concern manufacturing boilers generally confined its output to a single arrangement of boiler, hence that arrangement was known by the maker's name. Now that the primary patents have expired and many concerns have consolidated, most of the manufacturers will now make almost any type of boiler design; hence the names of the builders have in many cases lost their significance as to the types of boilers built by them.

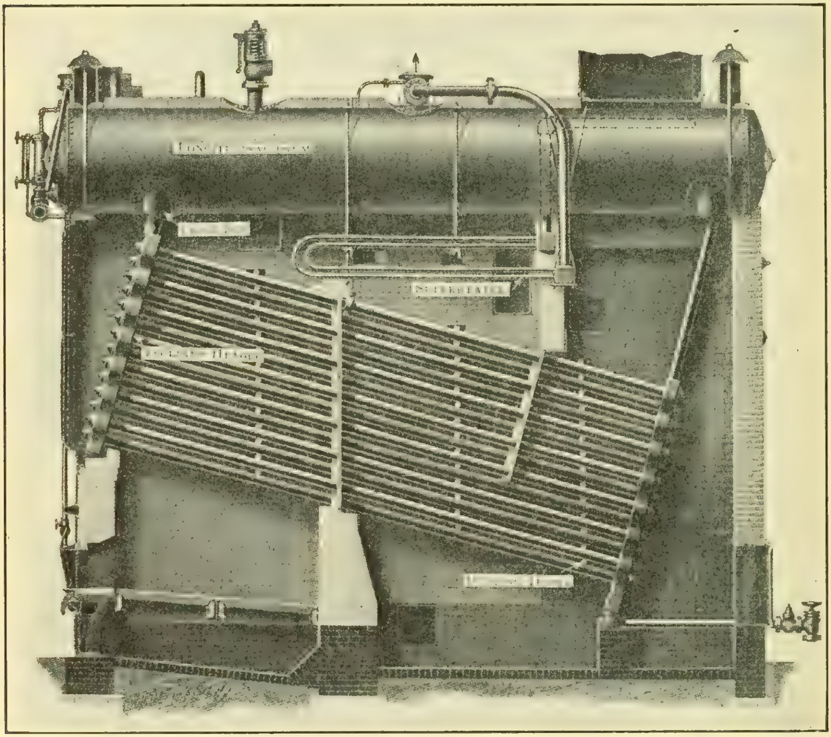


FIG. 725.—Elements of the B. & W. Type of Boiler.

staying. The headers are sinuous, so that the tubes are staggered vertically to baffle the gas flow; and opposite each tube is a handhole opening for cleaning or replacing the tube. Either *inclined headers*, as in Fig. 725, or *vertical headers*, Fig. 726, may be used. In the former the sloping tubes enter the headers at right angles, and in the latter they enter obliquely. The vertical headers, shown in Fig. 727, are the more expensive ones to make, but their use reduces to a minimum the overall length of boiler and drum obtainable with given length of tubes. In the vertical headers the handholes are elliptical and are closed, as shown in Fig. 728(*b*), by inside covers which are held tight both by means of the clamps and bolts and by the steam pressure, thin gaskets being interposed between the covers and their seats; and in the inclined headers the handholes are either the same as those in the vertical headers, or else are made circular, Fig. 728(*a*), in which case they are closed on the *outside* by means of forged steel caps which are held in place by means of clamps and bolts, the joints being metal-to-metal. The tubes used in these boilers are commonly 4 in. O.D. and from 18 to 20 ft. long; and they are often divided into two nests or decks, with interdeck superheaters between, the headers being arranged in series, as in Fig. 733.

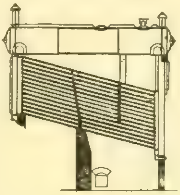


Fig. 726.—Vertical Header Arrangement.

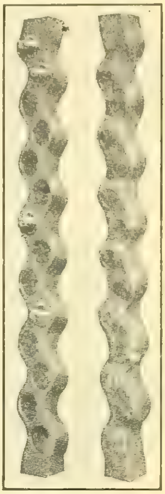
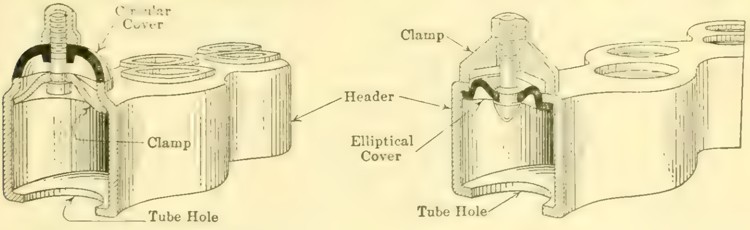


Fig. 727.—Vertical Headers.



(*a*) Header with Circular Handholes. (*b*) Header with Elliptical Handholes.

Fig. 728.

The arrangement of the front portion of a B. & W. boiler is shown in Fig. 729 with parts labeled. It will be noted that the relatively cold feedwater is injected into the body of the water in the drum and in the direction of flow; also that a steam baffle is introduced to prevent spouting from the front nipples, which would cause the entrainment of an undue amount of moisture in the steam released.

Boilers of this type ordinarily range in size from 750 to 10,000 sq. ft. of heating surface, the larger ones having two or three steam drums which are interconnected. Since they are of the "sectional type," i.e., built up of many small elements, the danger of disastrous explosions is very slight, as ruptures seldom occur elsewhere than in the relatively small tubes. The maximum pressure which it is ordinarily feasible to use with boilers of this type is about 450 lb. per sq. in., other types generally being more suitable for applications with higher pressures. The gases are usually baffled in such manner that they make three passes across the tubes, but other arrangements of baffling are sometimes used. Doors for external cleaning must be provided in at least one side wall of the setting, hence these boilers cannot be arranged in continuous batteries; instead, they are grouped in batteries of two each. Settings for these boilers and their furnaces were discussed in Sect. **460** (**c**).

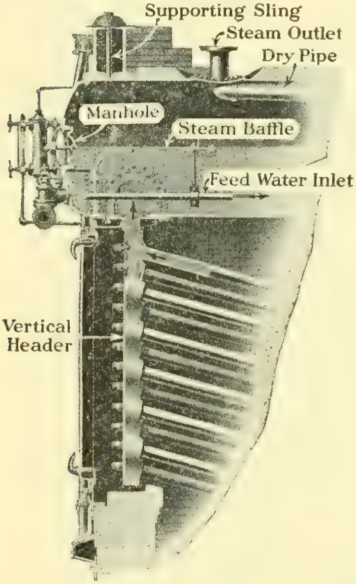


Fig. 729.—Front Section—B. & W. Boiler.

(**c**) The **box-header type** of longitudinal-drum boiler, which formerly was known as the **Heine type,** is now built by many firms and is extensively used because of its comparatively low cost. One arrangement of Heine boiler, shown in section in Fig. 730, consists primarily of the sloping steam drum and tubes, and the connecting headers at right angles thereto. The headers have individual handholes opposite the ends of each tube, and their large flat plates are strengthened by means of stay-bolts which are made hollow so as to permit the insertion through them of small steam or air pipes for blowing off the soot and dust from the exterior of the tubes. To compensate for the shell metal cut away at the large junction between the header and the drum, forged-steel *throat-stays* are used in the manner shown in Fig. 731(A) Large boilers of this kind may have more than a single drum; and inside these drums are located steam deflection plates and precipitation drums or " mud drums," as in Fig. 730. The feedwater is introduced into the mud drum and there remains quiescent and becomes heated to the temperature at which some of the impurities precipitate, before it mixes with the water which is circulating through the boiler. The deposits in the mud drums can be blown off at intervals through valves provided for the purpose. The main handhole plates are round and are inside-seating, with gaskets to seal the joints. A few large elliptical holes are provided for inserting the plates, which may be of the kind held in place by means of yokes and bolts (*B* in Fig. 731), or they may be " Key " caps (*C* in Fig. 731), which are rolled into slightly tapered holes and are held tight by the pressure within the boiler.

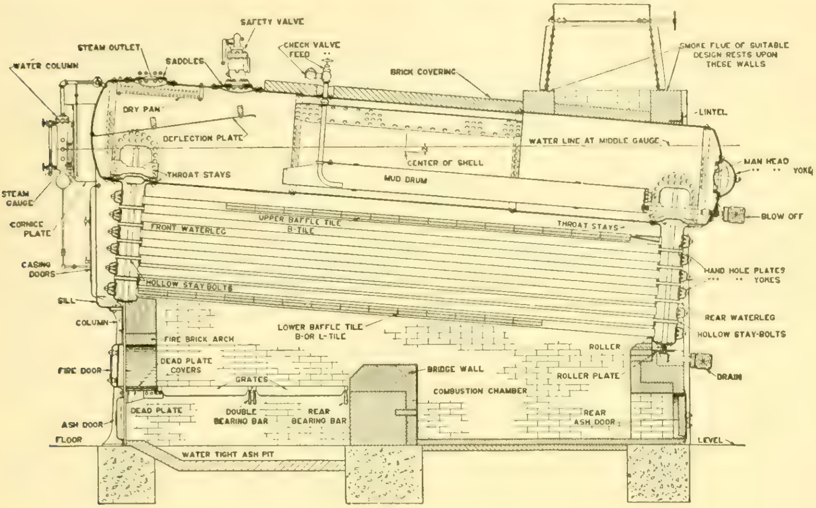


Fig. 730.—Section of Heine Boiler and Setting, with Names of Parts.

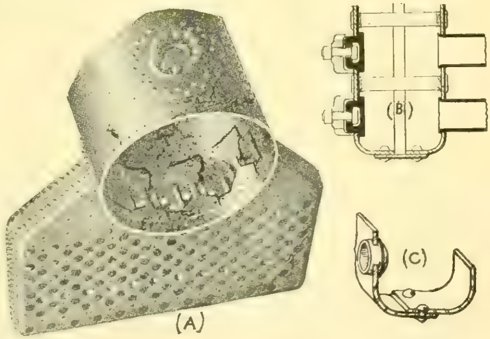
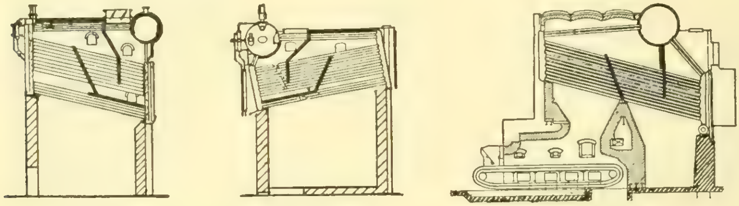


Fig. 731.—Junction of Heine Header and Drum. Handhole Caps.

These boilers are transported, in the smaller sizes, as completed units; in the larger sizes, the drums, tubes, headers, and other parts are shipped knocked down and are assembled at the power plant. The front of the boiler is either supported on cast-iron columns, as in Fig. 730, or is hung from above, and the rear water leg ordinarily rests on rollers which bear on cast-iron plates set in the top of the back wall of the setting, as shown in the same figure. With the longitudinal gas-baffling commonly employed with these boilers, no cleaning doors are required in the side walls of the settings; hence the boilers can be arranged in continuous batteries.

(d) **Other box and sectional-header boilers** differ from the ones described, in ways such as the following: the steam drum may be horizontal, instead of inclined; the mud drum or its equivalent may be differently arranged or be omitted; the tubes may be smaller, and one handhole may serve two or more tubes; instead of the tube and handhole sheets of the water leg being flanged and riveted in the manner shown in Figs. 730 and 731, the two sheets may be joined in other ways; and instead of longitudinal baffling, cross baffling may be used.

476. Horizontal Cross-Drum Water-Tube Boilers.—(a) This type of boiler was developed originally for marine applications where the headroom is limited; later it was adapted to stationary service and is now applied extensively in central stations in forms similar to those shown in Fig. 732. It consists fundamentally of straight longitudinal tubes, a single steam drum arranged transverse thereto, headers, and the necessary connecting nipples or tubes; and, except for the position of the drum, it is quite similar to the longitudinal-drum type of boiler just considered. The cross drum can be set lower, however, than is possible with the longitudinal drum, and, as seen from Fig. 732, it may be located either at the front or rear of the unit, or between these points. Its length corresponds to the boiler width, and may be 30 or 40 ft. in large units. As in the longitudinal-drum boilers, the headers may be either sectional or box, and the discharge from the circulating tubes con-



(*a*) Rear Drum. Sectional Header.

(*b*) Front Drum. Box Header.

(*c*) Intermediate Drum. Box Header.

Fig. 732.—Cross-Drum Boilers.

necting them to the drum is baffled to aid in separating the water from the steam. The tubes range in size from 2 to 4 in. O.D. Front or rear drums may be supported at their ends by chairs or columns, or they may be hung from above by suspension rods or straps; and, since they serve as beams from which the headers are suspended, they are made without circumferential joints except those at the ends. Headers not suspended from the drums may be either hung from overhead beams or be supported flexibly from underneath.

(**b**) Boilers of this type are limited to about 12,500 sq. ft. of heating surface when box headers are used. With sectional headers, the largest unit to date has 35,462 sq. ft. of surface with tubes 24 ft. long, arranged 51 tubes wide and 24 high, and with a drum 60 in. in diameter.[6] Since cross-drum boilers have but one drum, the cost is slightly less than that of longitudinal-drum boilers in sizes which would require the use of two or more drums with the latter; and they are the more feasible ones to use with the higher pressures (i.e., in the 700- and 1400-lb. classes) and with the higher rates of operation.

(**c**) **Examples** of the designs offered by one manufacturer of cross-drum sectional-header boilers for working pressures up to 500, 700, and 1450 lb. per sq. in., respectively, are given by the boiler elements shown in Figs. 733, 734, and 735.[7] In each of these units the boiler is provided with a slag screen, and each has water-cooled furnace walls (the first with water-cooled bottom and the other two with slag-tap bottoms), a convection-type superheater, and a tubular air heater.

In the **500-lb. unit,** Fig. 733, the three-pass boiler has a relatively extensive heating surface and accomplishes most of the heat absorption performed by the unit. It is followed only by an air preheater. The boiler tubes are arranged in two nests or banks, with space between for an interdeck superheater. The lower nest of tubes is five rows high, and the upper one fifteen rows high, the respective headers being arranged in series. The boiler tubes are 4 in. O.D. and 22 ft. long; and the drum is of riveted construction. Data regarding physical and approximate performance characteristics of such a unit designed for a drum pressure of 420 lb. per sq. in. and for delivering steam at a temperature of 800 deg. fahr. are given in Table LXIII.

In the **700-lb. unit**, Fig. 734, the single-pass boiler has a relatively small proportion of heating surface, as it is followed by a steaming economizer and a tubular air preheater, both of which use the counter-flow principle. The tubes are $3\frac{1}{2}$ in. O.D. by 20 ft.; each section is six rows high, and the drum has riveted joints. Data are given in Table LXIV.

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[6] Power Plant Engineering, July 1, 1928, p. 716.

[7] Proc. N.E.L.A., 1931, pp. 780–789.

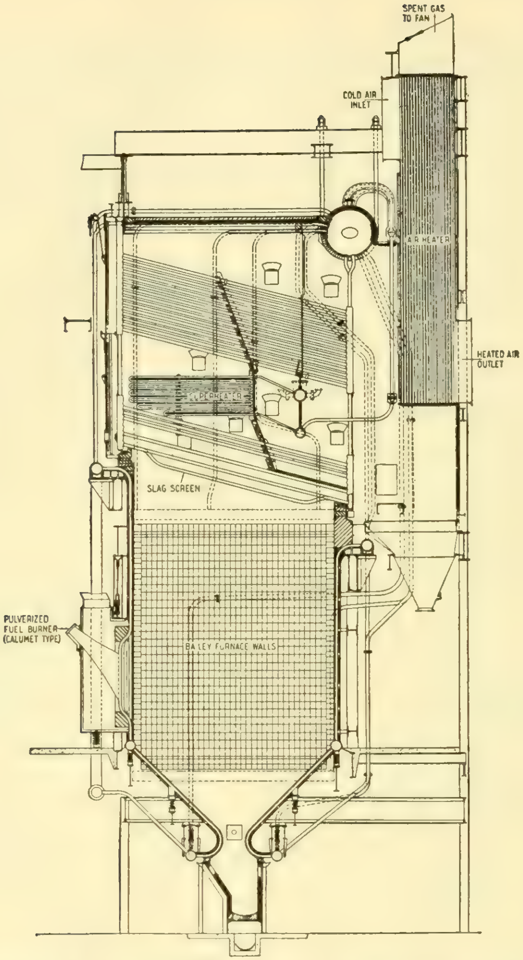

Fig. 733.—Typical B. & W. Cross-Drum Unit for Pressures up to 500 Lb. per Sq. In.

TABLE LXIII

Physical and Approximate Performance Characteristics of 420-Lb. Unit.

Lb. steam per hr. per ft. width of furnace	11,000
Lb. flue gas per hr. per ft. width of furnace	14,500
Lb. air per hr. per ft. width of furnace through air heater	12,500
Boiler heating surface per ft. width of furnace, sq. ft.	925
Superheater surface per ft. width of furnace, sq. ft.	275
Air heater surface per ft. width of furnace, sq. ft.	2,100
Boiler Tubes: 4 in. outside diam. × 22 ft. long. 15 rows high in upper bank. 5 rows high in lower bank.	
Drum pressure, lb./sq. in.	420
Steam Temperature, F.	800
CO_2 in gas leaving boiler, per cent	15
Temperature of gas leaving boiler, F.	775
Temperature of gas leaving air heaters, F.	398
Temperature of feedwater, F.	330
Temperature of air to air heater, F.	80
Temperature of air leaving air heater, F.	550
Combined efficiency of boiler, superheater, air heater, and furnace, per cent	84.8
Draft loss through boiler, in. water	3.83
Draft loss through air heater, in. water	1.45
Air resistance through air heater, in. water	2.20
Pressure drop, drum to superheater outlet, lb. per sq. in.	30

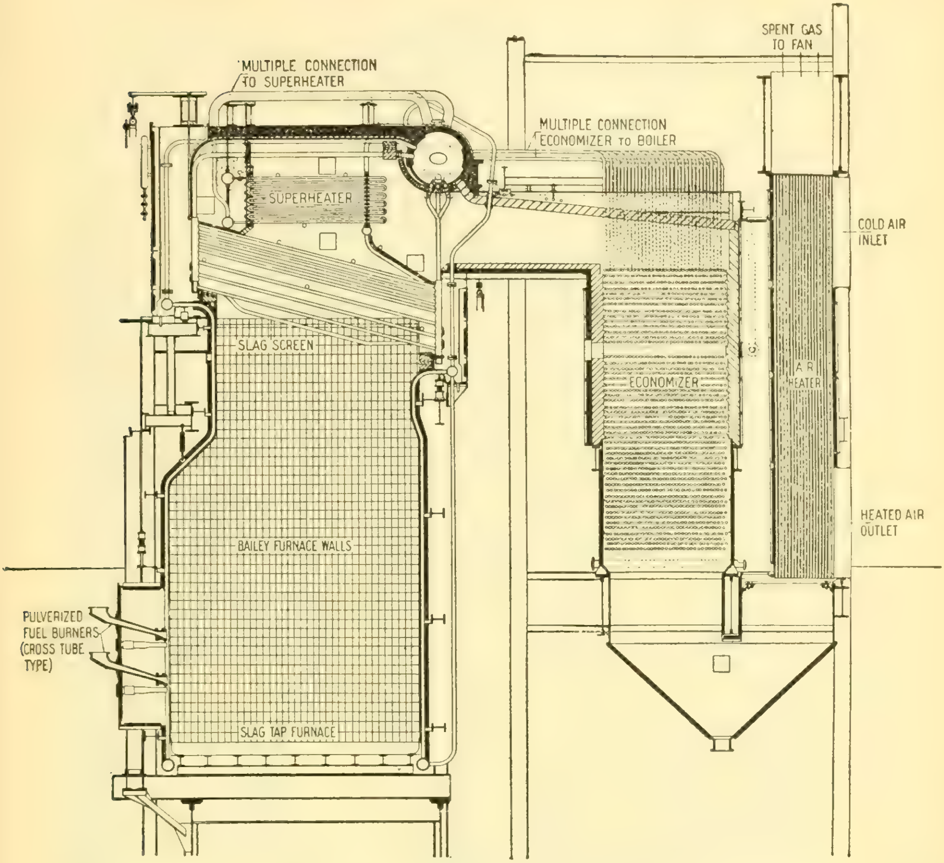

FIG. 734.—Typical B. & W. Unit for Pressures up to 700 Lb. per Sq. In.

TABLE LXIV

PHYSICAL AND APPROXIMATE PERFORMANCE CHARACTERISTICS OF A 650-LB. B. & W. UNIT

Lb. steam per hr. per ft. width of furnace	13,600
Lb. flue gas per hr. per ft. width of furnace	19,000
Lb. air per hr. per ft. width of furnace	14,750
Boiler heating surface per ft. width of furnace, sq. ft.	270
Superheater surface per ft. width of furnace, sq. ft	220
Economizer surface per ft. width of furnace, sq. ft.	785
Air heater surface per ft. width of furnace, sq. ft.	1,750
Boiler Tubes: Diameter $3\frac{1}{4}$ in. O.D., length 20 ft. Six rows high.	
Steam-Pressure at superheater outlet, lb./sq. in	650
Temperature at superheater outlet, F.	760
Temperature of gas leaving economizer, F.	652
Temperature of air to air heater, F.	80
Temperature of air from air heater, F.	486
Temperature of gas from air heater, F.	356
Temperature of feed to economizer, F.	350
Efficiency of boiler, superheater, air heater, and furnace, per cent	83
Draft loss through boiler, superheater, and economizer, in. water	6.2
Draft loss through air heater, in. water	3.2
Air resistance through air heater, in. water	3.1
Pressure drop through economizer, lb./sq. in	20
Pressure drop through superheater, lb./sq. in	40

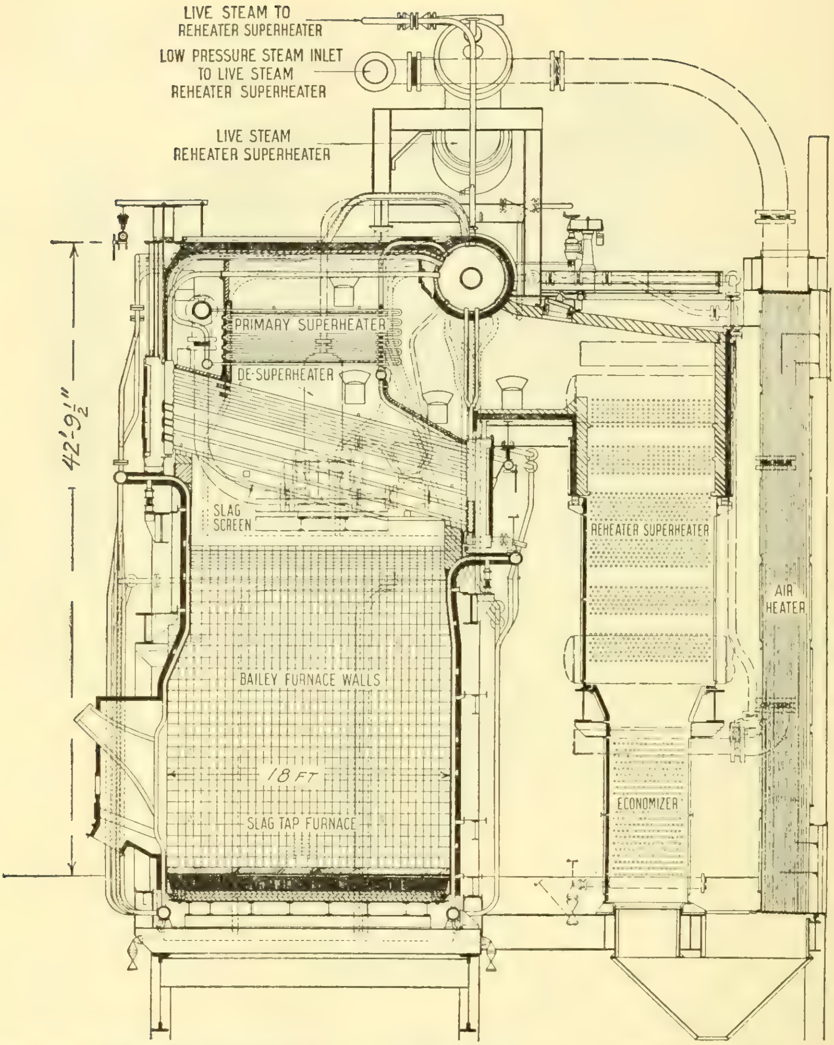

Fig. 735.—Sectional Side Elevation of 1450-Lb. Babcock & Wilcox Installation at South Amboy Station of Jersey Central Power & Light Company.

TABLE LXV.—Physical Data for 1450-Lb. B. & W. Unit

	Sq. Ft.
Boiler—40 tube-sections wide; drum, 52 in. I.D., solid forged	7,412
Superheater, with desuperheater to give a steam temp. of 750 deg. fahr.	12,900
Gas Reheater—Tubes, 2 in. O.D. × 24 ft. 6 in. long, arranged 24 wide × 42 high, with steam making 6 passes through 3 loops of 7 tubes each	12,900
Live Steam Reheater—42-in. drum—262 tubes 2 in. O.D. × 16 ft.	2,269
Economizer—Tubes 2 in. O.D. × 24 ft. long, arranged 16 wide × 22 high	4,423
Air Heater—Tubes 2½ in. O.D. × 42 ft. long, arranged 22 rows of 53 each	32,052

In the **1450-lb. unit,**[8] Fig. 735, the boiler is quite similar to that in the preceding figure, except that it has eight 3¼-in. tubes 20 ft. long in each section, and a solid-forged drum. In addition to the boiler, furnace, and superheater, this unit contains a desuperheater, a convection-type reheater, a live-steam reheater, and economizer, and an air heater. The superheater is proportioned to deliver steam at 750 deg. fahr. at minimum load, and at higher temperatures at greater loads; but a thermostatically controlled tubular desuperheater maintains the final steam temperature always at 750 deg. fahr. The convection-type reheater is planned to deliver the steam at 750 deg. fahr. under maximum load. Under lighter operation this same temperature is maintained with the aid of a thermostatically controlled live-steam reheater of the tubular type. Data for a 1400-lb. unit similar to this one are contained in Tables LXV and LXVI. The reheaters of this unit are discussed in Sect. **489.**

(d) **Single-pass cross-drum boilers** of interest are shown in the units illustrated in Figs. 736 and 737. Special units having such arrangements are of comparatively simple construction, occupy minimum floor space and building volume, can be forced to relatively high capacities, and, being without baffles, can operate solely with natural draft or with such draft assisted with induced-draft fans requiring but little power to drive them.

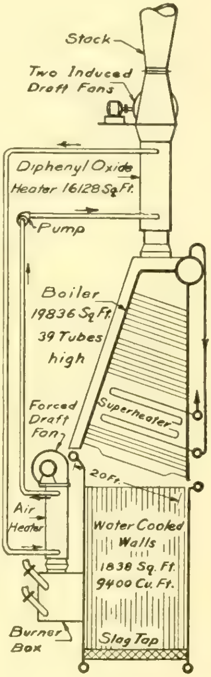

FIG. 736.—Single-Pass B. & W. Boiler. (Bremo.)

The unit shown diagrammatically in Fig. 736 is located in the Bremo Station of the Virginia Public Service Corporation [9] and is designed to deliver steam at 525 lb. per sq. in. pressure and 825 deg. fahr., to have a normal output of 160,000 lb. per hr. with maximum capacity of 200,000 lb. per hr., and to serve by itself a 15,000-kw. turbine. In this arrangement the sectional headers of the boiler form a tapered gas passage of such convergence that the gases maintain a nearly constant velocity while flowing through the boiler, despite their decrease in temperature. No economizer is used, but an air-heater system of novel design is included, in which the air heater

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[8] For a description of this type of unit, see Power Plant Engineering, July 15, 1930, p. 788; also Power, May 28, 1929, p. 863.

[9] Bremo: "An Outstanding Central Station," Power Plant Engineering, May 1, 1931, p. 486.

TABLE LXVI

Performance Characteristics of 1450-Lb. B. & W. Unit at Various Rates of Output (South Amboy Station).

Pounds of steam per hour	99,140	158,462	203,230	248,370	273,000
Drum safety valve pressure, pounds per square inch	1,400	1,400	1,400	1,400	1,400
Superheated steam temperature, F.	750	750	750	750	750
Per cent CO_2 at damper	15	15	15	15	15
Per cent CO at damper	0	0	0	0	0
Temperature of flue gas leaving economizer, F.	522	585	612	635	650
Temperature of flue gas leaving air heater, F.	250	285	303	317	325
Temperature of water entering economizer, F.	405	450	460	498	505
Temperature of water leaving economizer, F.	431	478	506	522	528
Temperature of air entering air heater, F.	80	80	80	80	80
Temperature of air leaving air heater, F.	420	455	466	477	486
Efficiency of boiler, superheater, reheater, economizer, and furnace, per cent	81.7	80.1	79.5	78.9	78.5
Efficiency of boiler, superheater, reheater, economizer, air heater, and furnace, per cent	88.3	87.5	87.0	86.7	86.5
Draft loss through boiler, reheater, and economizer, inches of water	0.90	2.00	3.00	4.20	5.00
Draft loss through air heater, inches of water	0.75	1.70	2.50	3.45	4.10
Air resistance through air heater, inches of water	0.50	1.15	1.70	2.35	2.80
Pounds of flue gas per hour	126,000	191,000	233,000	275,000	300,000
Pounds of air through air heater per hour	106,000	160,000	196,000	231,000	252,000
Gas Reheater:					
Pounds of steam per hour	85,000	134,000	169,000	204,000	223,400
Steam pressure, pounds per square inch absolute	161	251	316	385	420
Heat content of steam entering, B.t.u. per lb.	1201.5	1219.5	1236.0	1253.0	1256.0
Steam temperature leaving reheater, F.	750	750	750	750	750
Steam pressure drop through reheater, pounds per square inch	3.0	4.8	6.1	7.3	8.0

is located immediately adjacent to the powdered-coal burners and receives its heat from a liquid substance which is 85 per cent diphenyl oxide and 15 per cent naphthylene. This liquid is circulated through the air heater and a heat-absorbing element that is located between the boiler outlet and the stack, as shown in the figure. This arrangement avoids the necessity of using long, expensive, and voluminous air ducts which would offer considerable resistance to flow, and would require relatively large power for the forced-draft fans.

The unit shown in Fig. 737 is installed in the **Duluth** Steam Electric Station, which serves only as a " standby " to a hydro-electric system. It generates steam at 485 lb. per sq. in. and 760 deg. fahr., has a maximum continuous steam-generating rate of 300,000 lb. per hr., and by itself serves the demands of a 25,000-kw. turbine. The height of the boiler section (43 tubes high) is sufficient to eliminate the need for an economizer or air heater; and no induced-draft fans are required to supplement the stack draft because the "chimney effect" of the boiler itself entirely overcomes the resistance to gas flow through the unit except at the higher loads. It is claimed that in large sizes this arrangement is economically justified for use in standby plants because of its low operating and maintenance cost and the saving effected by it in building volume.[10]

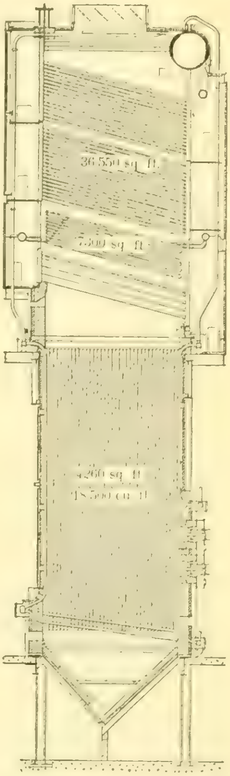

FIG. 737. — Single-Pass Walsh-Weidner Boiler. (Duluth.)

477. Vertical and Semi-Vertical Straight-Tube Water-Tube Boilers.—(a) Typical boilers of this class are illustrated in Figs. 738, 739, and 740. The first figure shows the **Wickes** type of vertical water-tube boiler, which consists of single upper and lower drums and straight vertical tubes connecting them. The boiler itself is of simple design, low in first cost, and occupies little space, but it requires the use of a Dutch oven type of furnace; and for tube removal and insertion, which is done through handholes located in the dome of the steam drum, it is necessary to provide adequate room, or a skylight, immediately of above it. Such installations are limited to moderate rates of operation, and are built in sizes up to 5000 sq. ft. of heating surface and for pressures up to 200 lb. per sq. in.

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[10] "Single-Pass Boiler Installation at Duluth Station," by A. H. Krauss, Combustion, March, 1932. Also, "The Single-Pass Boiler and Its Economic Field," by O. de Lorenzi, Power Plant Engineering, Aug. 15, 1931, p. 837.

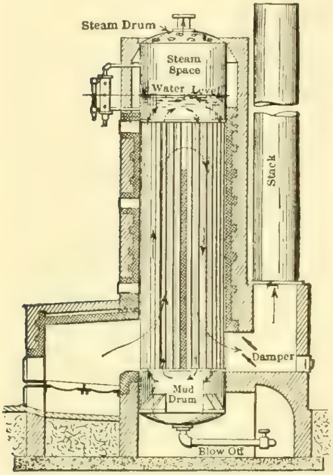


Fig. 738.—Wickes Type of Vertical Water-Tube Boiler.

(b) The **Bigelow-Hornsby** semi-vertical boiler of this class, Fig. 739, consists primarily of a number of simple vertical and oblique elements—each composed of an upper and a lower drum of small size and a bundle of 21 straight tubes connecting them—a single horizontal steam drum, and circulation tubes connecting the drums. The elements are assembled at the factory, and they and the other parts are shipped knocked down. The boilers are suspended freely from above, and their baffles are arranged merely to fill the spaces between the elements and to give straight-through single-pass flow to the gases. These boilers are built in sizes ranging from 3750 to 30,000 sq. ft. of heating surface.

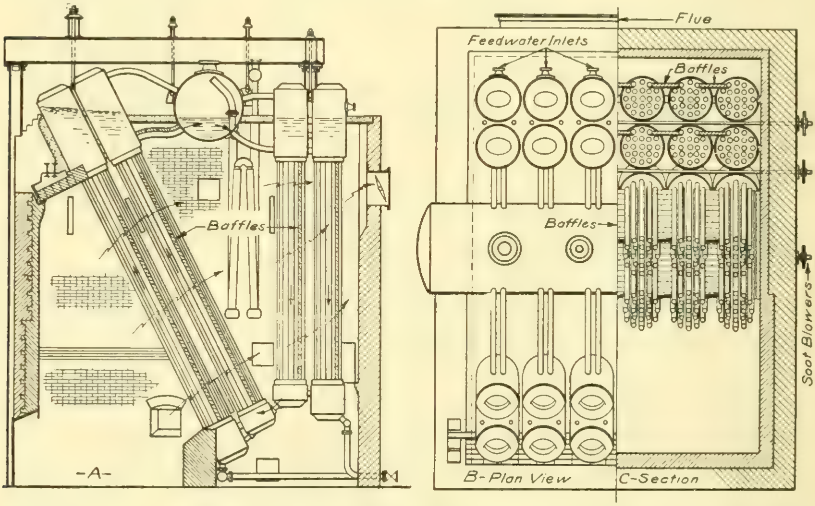


Fig. 739.—One Arrangement of Bigelow-Hornsby Semi-Vertical Water-Tube Boiler.

(c) The **Edge Moor single-pass** unit shown in Fig. 740 is also of this class. It consists of a heating section, a steaming section, and a superheater. The heating section is composed of straight tubes assembled in short, rectangular, cast-steel boxes and arranged to form a sinuous path for the flow of the water. The steaming section is made up of a number of elements, each consisting of small upper and lower drums with connecting tubes (the axes of the drums being at right angles to the tubes), a steam drum arranged crosswise, steam tubes between the upper drums, and drain tubes from the steam drum to the lower ones. No baffles are used in this unit; and the gas flow is in a direct and unobstructed path and has a practically uniform velocity throughout the transit because the number of tubes and spaces between them are ranged to form a passage with width decreasing in accordance with the reduction in specific volume of the gases which accompanies their decrease in temperature. The feedwater enters the unit at the rear (at *A*), follows a sinuous path from tube to tube in the heating section in a general direction counter-current to the flow of the gases, and is discharged into the main cross drum (1) at saturation temperature. The steaming section merely performs the function that its name indicates.

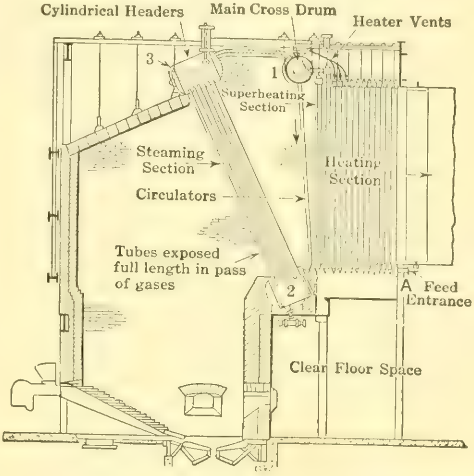
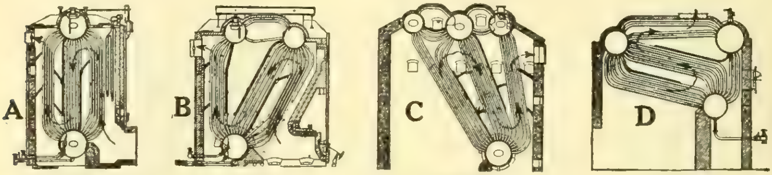


Fig. 740.—Edge Moor Single-Pass Boiler.

478. Bent-Tube Water-Tube Boilers.—(a) Boilers of this type are primarily composed of simple curved tubes and drums which, in the larger units, are not assembled until they reach their destination. Boilers of such construction can be built in sizes ranging from very small ones to the largest so far installed; and, by varying the number and sizes of the drums, and the lengths, number, sizes, and arrangement of the tubes, the designs of such boilers can be readily modified not only to harmonize with the proportions of their furnaces and other related parts, but also to conform to the needs of a wide variety of operating and economic conditions. To meet the diverse requirements encountered, each manufacturer of boilers of this type makes them in a number of different forms. For example, some of those made by one concern are shown in Fig. 741; the relatively inexpensive two-drum arrangement may be adopted where a surplus of waste heat is available; the four-drum design is suitable for a moderate-pressure fuel-burning power-unit, the three-drum arrangement for a high-pressure unit requiring the use of costly solid-forged drums, and the low-head, semi-horizontal structure where headroom is limited. The drums are the most expensive elements of these boilers, and the cost of each depends on its size and the thickness of its shell, the latter depending, in turn, on the pressure for which the drum is designed and on the widths of the ligaments of metal left between the tube holes in the longitudinal rows drilled in the shell. Thus, the number and diameter of the drums, and the arrangement of holes in them, are factors of considerable economic importance. In general, bent-tube boilers, especially when used with



WASTE HEAT BOILER — LADD 3 Drum — HEINE V-Type Boiler — HEINE VM-Type

FIG. 741.—Some Typical Forms of Bent-Tube Water-Tube Boilers (as made by The Combustion Engineering Corporation).

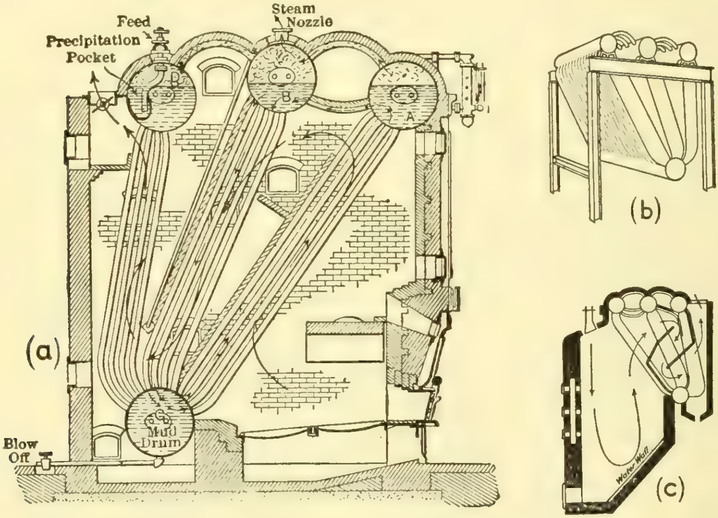


FIG. 742.—Stirling Type of Boiler.

moderate pressures, cost less than water-tube boilers of the types composed of the numerous elements of complicated form.

All drums of such boilers have manholes which give access to their interiors and to the tube ends, hence the necessity of having a multiplicity of handholes is avoided. The upper drums are rigidly supported by external structural framework, and the tubes and lower drum are freely pendent therefrom. The side walls of the settings must be provided with cleaning doors exposing the exteriors of the tubes, hence these boilers cannot be arranged in continuous batteries—however, they may be grouped in pairs.

(**b**) An *early form* of bent-tube boiler was the **four-drum arrangement** shown in Fig. 742, made by the former Stirling Company; the name **Stirling type** has been applied quite commonly to all the widely used four-drum boilers of that or similar design. Boilers of this general type are used extensively with moderate pressures and are now manufactured by many concerns. In the arrangement shown in Fig. 742, the feed-water enters a precipitation pocket in drum *D* and is heated there and as it descends to the mud drum *C*. The circulation of the water is through the circuit formed by the tubes joining the drums *A*, *B*, and *C*. All the upper drums are connected by steam pipes, and they rest on beams that are supported by steel columns, as in (*b*) in Fig. 742. The rows of tubes running circumferentially around the drums are arranged in pairs, between which is sufficient space to permit of the removal or insertion of tubes located in the interior of the nests. Space must be provided above or underneath the boiler for handling the tubes when making renewals. The gases may be baffled longitudinally with respect to the tubes, as in (*a*) in the figure, or cross-baffled as in (*c*). Some modifications of the tube arrangements of these boilers, as introduced by various concerns, are shown in Figs. 741(*C*) and 743.

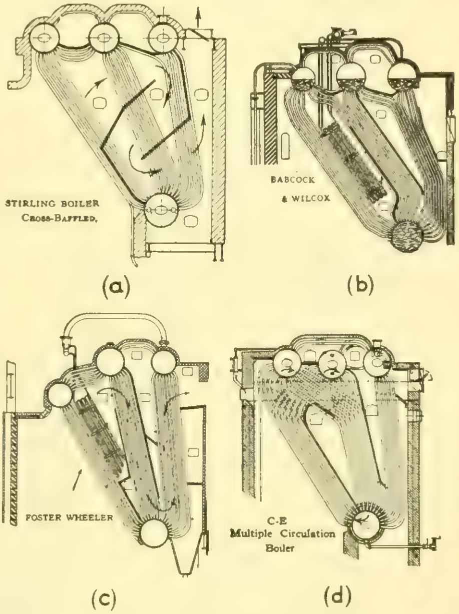


Fig. 743.—Some Tube Arrangements of Four-Drum Bent-Tube Boilers.

(c) The **three-drum boilers** have less steam-drying space in their drums than is provided in the four-drum arrangement, hence are often limited to more moderate rates of operation than are the latter, unless used with high pressures. When such a boiler is to be operated intensively it is usually supplemented by an integral economizer, such as *E* in Fig. 744, and perhaps also by an air heater. Sometimes a supplementary steam-drying drum is added to insure furnishing dry steam to the steam mains or superheater; and, if the connections to this drum are exposed to the hot gases, the drying process is aided thereby and a slight superheat may even result. The three-drum arrangement has been used frequently for high-pressure installations, one example of which is given in Fig. 807 (p. 756), which shows a 1400-lb. unit at Lakeside, Milwaukee. When all three drums have the same number of tubes between them, as in Fig. 745, the boiler is commonly referred to as having the **ring circuit arrangement.**

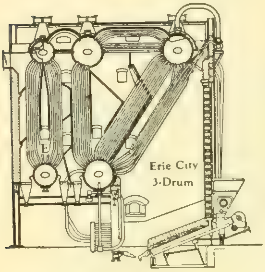


FIG. 744.

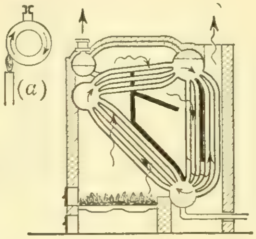


FIG. 745.

(d) The so-called **Combustion Steam Generator,**[11] shown in Fig. 746, is a specialized type of unit built around the furnace, its two-drum bent-tube boiler, or "convection element," being only a minor part. This design combines in one structure all the elements required for

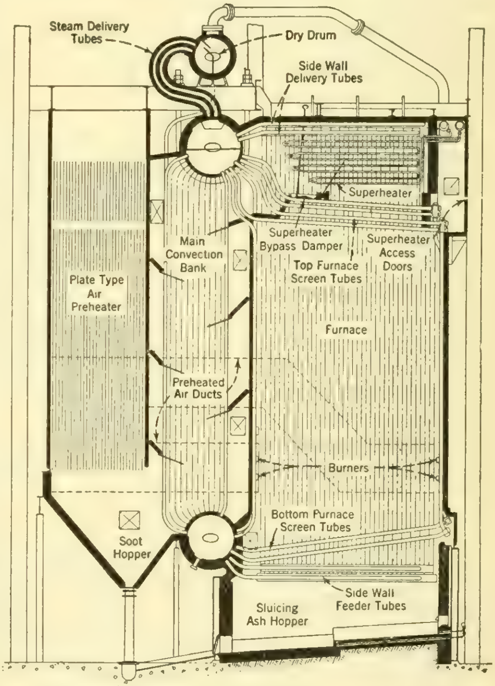


FIG. 746.—Combustion Steam Generator.

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[11] Trade name adopted by the Combustion Engineering Corporation for this arrangement of unit.

fuel-burning, steam-generation, superheating, and ash disposal. The furnace is completely enclosed in water walls which absorb over 60 per cent of the heat transmitted to the unit; it is tangentially fired from the corners by means of pulverized-coal, gas, or oil burners located in the lower part of the furnace, and is designed for a heat liberation of 30,000 B.t.u. per hr. per cu. ft. when using an average grade of coal. The furnace walls absorb heat at the rate of about 37,500 B.t.u. per hr. per sq. ft., and the convection surface at about one half this rate. A dry-steam drum is placed above the steam-and-water drum, being interposed between the latter and the superheater. A superheater by-pass damper affords means for controlling the final steam temperature. This very compact unit is enclosed in an air-tight and dust-tight casing which is free from openings.[12]

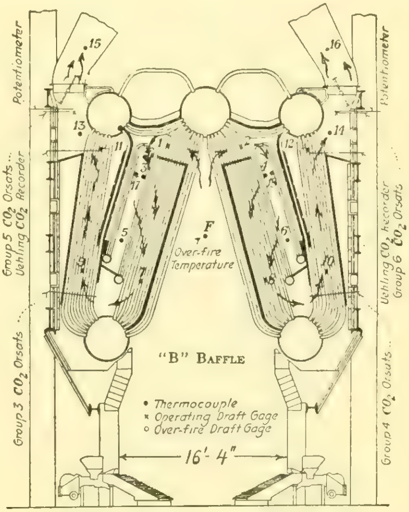

FIG. 747.—Type-W Stirling Boiler.

479. Large Double-Set Units.—(a) Very large steam-generating units can be formed by setting over a single furnace two semi-vertical, vertical, or horizontal water-tube boilers similar to any of the usual commercial ones already illustrated.

(b) The **Type-W Stirling boiler** (B. & W.) shown in Fig. 747 is essentially a combination of two three-drum boilers with one drum in common. Many of these boilers, the first with double settings made in this country, are used in the plants of The Detroit Edison Co., that shown in the figure having 23,654 sq. ft. of heating surface and being stoker fired from both sides. The best arrangement of baffles in this unit was determined as a result of fifty very extensive tests and is that shown in the figure.[13] Similar units, except that they are fired by pulverized coal, have been installed in the Trenton Channel Plant of this company.[14]

(c) The **East River Unit** shown in Fig. 748 is the largest so far constructed, and several like it have been installed in the East River Plant of the New York Edison Co.; both of its double-set Ladd boilers are of the three-drum type to which have been added dry drums and integral economizers, the latter formed by the outer nests of tubes. The unit is fired in the manner shown in Fig. 609, p. 531, and has been operated up to an output of 1,270,000 lb. of steam per hour. The extent and distribution of the heating surface in this unit, expressed in square feet, are as follows: boiler, 60,706; furnace, 7345; superheater, 13,900; air preheater, 82,720; economizer, 15,366. Other data are: test pressure, 750 lb. per sq. in.; working pressure, 425 lb. per sq. in.; steam temperature, 725 deg. fahr.; furnace volume, 38,200 cu. ft.; furnace operating range from 10,000 to 50,000 B.t.u. per hr. per cu. ft.; efficiency of the unit, 88 per cent with an output of 500,000 lb. steam per hr., and 86.5 per cent at 1,000,000 lb. per hr.

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[12] For additional description and test data see: "Steam Generation Unit," by Henry Kreisinger, Power Plant Engineering, Nov. 1, 1931, p. 1053.

[13] "Tests of a Type-W Stirling Boiler at the Connors Creek Power House of The Detroit Edison Co.," by P. W. Thompson, Trans. A.S.M.E., 1922, p. 1015.

[14] For a very complete description of this plant, see Engineering (London) March 23, 1928, p. 341; for revamping of this plant see Power Plant Eng., Apr., 1935.

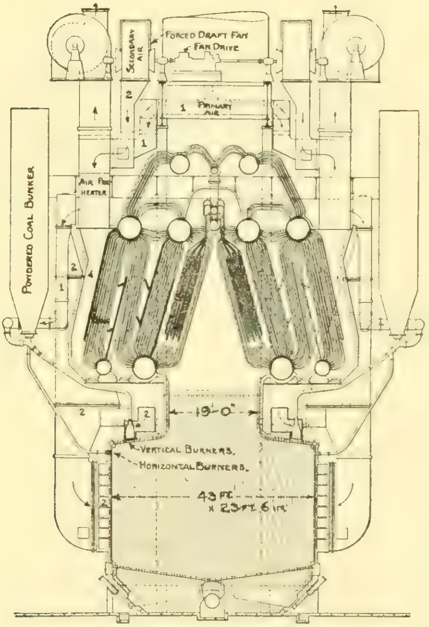

Fig. 748.—East River Double-Set Unit.

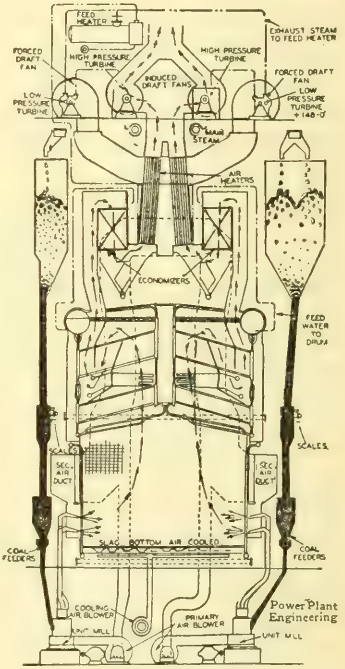

Fig. 749.

(d) The **Hell-Gate Double-Set Unit,** shown in Figs. 749 and 750, contains two contraposed B. & W. boilers of the horizontal cross-drum type, not interconnected, and is designed to generate 1,000,000 lb. of steam per hr. at 275 lb. per sq. in. pressure and at a temperature of 750 deg.

fahr. The heating surface of the unit is distributed, in square feet, as follows: boiler, 52,306; furnace, 4590; superheater, 12,000; economizer, 19,604; air heater, 60,500. The furnace has a volume of 45,100 cu. ft. and is designed for a liberation rate of 24,600 B.t.u. per hr. per cu. ft. when the unit is delivering 800,000 lb. of steam per hr. It has a slag-tap bottom, and is fired by twelve horizontal cross-tube burners, served by four mills. The bottom boiler tubes are carried around under the return sectional headers to give them protection and to support the tile under them and between the boilers.[15]

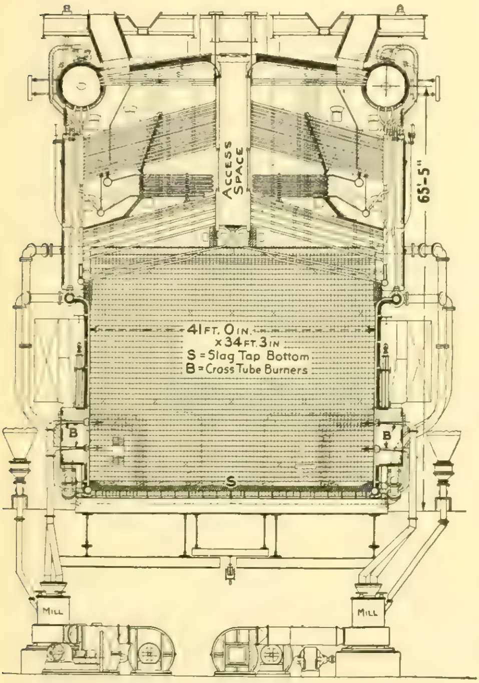

FIG. 750.—Hell-Gate Double-Boiler Unit.

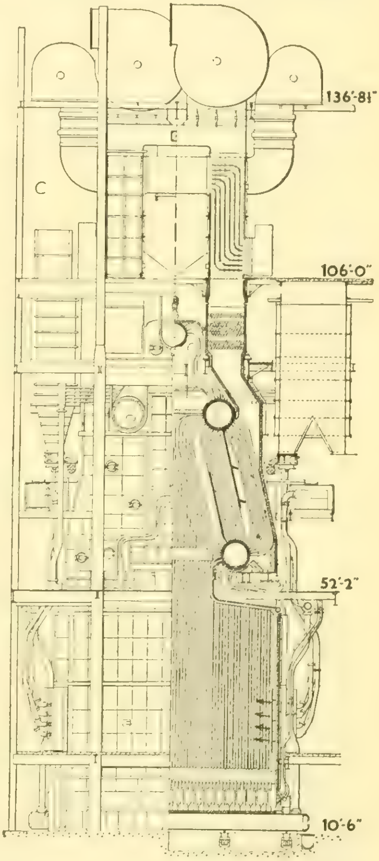

FIG. 751.—Kips Bay Unit (No. 5).

(e) The **Kips Bay unit,** shown in Fig. 751, contains two Ladd two-drum boilers and exemplifies the concentration of great steam-generating capacity per square foot of ground area in a location where land is expensive. This 300-lb. heating unit, capable of producing between 700,000 and 1,000,000 lb. of steam per hour, occupies an area only 29 ft. × 40 ft., but has a height of 145 ft., equaling that of a 12-story building. Its capacity is double that of previous boilers of the same floor space in this plant.[16]

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[15] This unit is installed in the Hell Gate Plant of the United El. Light and Power Co., New York City. See: "The Power Behind New York," Part III, Power Plant Engineering, Aug. 1, 1931, p. 799; "Double Cross-Drum Boilers," Power, March 3, 1931, p. 356.

(f) The **1400-lb. Ford units** of 700,000 lb. per hr. designed capacity, in Rouge Plant No. 1, have boilers similar to those at Kips Bay, but with solid-forged drums. These units are fired by a combination of pulverized coal and blast furnace gas, about 92,000,000 cu. ft. of the latter being available per day in this plant.[17]

480. Other Principles and Arrangements of Boilers.—(a) The desire to use higher steam pressures and temperatures, to operate the boilers more intensively and efficiently, to have greater ability to follow rapidly changing loads, to operate over longer periods between enforced shut-downs, and to reduce the cost of boilers and of the space occupied by them, has stimulated search for new and more useful principles and constructions applicable in designing boilers. It is the purpose of this section to give consideration to various arrangements and principles which are not generally encountered in the more usual forms of boilers as now built, but which may possibly be found of value in future practice. First, however, it is desirable to review briefly the more important features of the *conventional boilers* considered in the preceding section. Most of these boilers are of the *recirculation type* consisting of combinations of tubes and drums, often with headers, through which a considerable body of water traverses a continuous closed circuit and does so repeatedly under the action of *natural circulation* caused by the heat received from the furnace; and they all have drums which are pierced by numerous tube holes, to compensate for which requires the use of thicker walls, which adds to the expense of the apparatus. Also, the usual boiler receives most of its heat by convection instead of by the more intensive method of direct radiation, and its water is evaporated mainly while in contact with the heat-transmitting walls, a method which promotes the formation of scale on these walls if the water is not pure.

(b) In designing **boilers to meet the greater demands,** effort is made to insure at all times an adequate flow of water over all parts of the heating surface, to have the greatest possible proportion of the heat transmission take place by direct radiation, to apply the counterflow principle, to avoid scale deposition, and to simplify the structure and to make it safe. While in the United States the somewhat conventional boilers of the horizontal cross-drum type and of the vertical and semi-vertical type have been extensively used for pressures up to 1500 lb. per sq. in. and even above, many other high-pressure experimental or special designs of interest have also been developed in this country. In Europe many radically different and special designs have been offered to meet the high-pressure requirements, but most of them, however, are at present still in the development stage. Many of them avoid having the evaporation of the water take place in tubes that are directly exposed to the radiation from the furnace and make use of forced circulation or the equivalent; some depend entirely on uninterrupted automatic control of the water and fuel; in most of them the tubes are of quite small diameter and are very thin; in some designs the use of drums and headers is eliminated altogether, and in others the drums are simple cylinders which are free from holes and hence can be made relatively thin and at low cost. Some of the expedients which may be applied and some of the newer or less usual steam-generating systems of interest will be considered briefly in the remainder of this section.

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[16] For detailed description, see: "Fifth Boiler Unit now in Operation at Kips Bay Station of N. Y. Steam Corp.," by A. R. Ludt, Combustion, Jan., 1931, p. 30.

[17] Power Plant Engineering, March 15, 1931, p. 354; also Power, Jan. 6, 1931, p. 5.

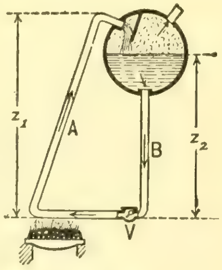

FIG. 752.

(c) **Definite circulation** may be insured by discarding reliance on natural convection currents and using instead some form of pumped or *forced circulation* with definite paths for the fluid to follow. Forced flow is an inherent requirement in boilers of the *once-through type,* in which the working substance in liquid form is injected into their tubes at one end and the steam formed is discharged from the other end into a steam drum or directly into the delivery pipe, there being no recirculation of the fluid.

The short-circuiting and reversal of flow, which may occur under unfavorable conditions in boilers having nests of tubes that connect common headers or drums and which deliver the mixture of liquid and steam below the surface of the water, may be avoided by having the tubes *discharge above the water level,* as shown diagrammatically in Fig. 752. Though the height z_1 in the figure is greater than z_2, positive flow can readily be maintained in the definite direction indicated by the arrows, the circulation being produced entirely by the proper application of the heat. Sometimes a non-return valve, V in the figure, is inserted to insure proper direction of flow, but after circulation is once established this valve is not needed, and often it is omitted altogether.

The use of tubes discharging above the water level in this manner, in place of those having submerged or "drowned" outlets, was introduced in 1887 by *Thornycroft* in marine boilers; in one early design of this feature, shown in Fig. 753, the downward circulation is through the large "down-comer" tubes located outside of the boiler casing, as shown. Incidentally, in this arrangement the tubes in the inner row are bent so as to fill the space between the tubes of the second row, thus forming a continuous surface over the furnace except near the lower drum where a gas passage is left. The two outer rows of tubes are bent in a similar manner to form an outer shell with opening only at the top.[18]

(d) Very rapid circulation, or so-called **accelerated circulation,** resulting in great steam-generating capacity, is obtained in the arrangement of marine boilers shown in Fig. 754 and known as the **express type of boiler,** in which the tubes are small, usually 1 in. O.D., and the ones serving as risers are exposed to the direct radiation from the fuel. These compact, light weight, " small-tube boilers " are very extensively used in torpedo boats and other light, high-speed vessels, but are suitable also for stationary practice.[19] Boilers of this general type are made by Thornycroft, Yarrow, Normand, B. & W., Foster Wheeler, and others, and are commonly designated by the names of their manufacturers.

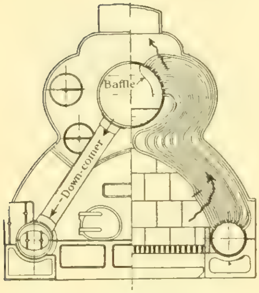

Fig. 753.—Early Thornycroft Boiler.

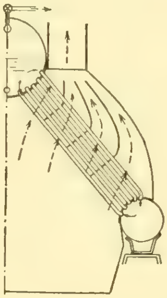

Fig. 754.—Express Type of Marine Boiler.

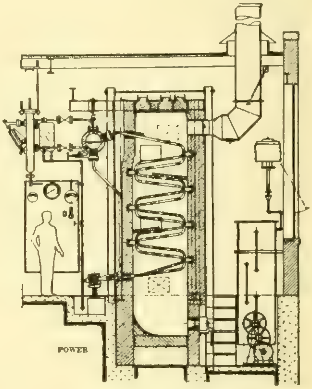

Fig. 755.—Yarnall-Waring 2500-Lb. Boiler.

(e) The recently constructed **Yarnall–Waring 2500-lb. test boiler** is also of the small-tube, rapid-circulation type. This small boiler,

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[18] The arrangement shown has the disadvantage of producing serious oxidization of the upper portions of the tubes when the boiler is shut down for long periods, for air pockets are likely to form in the upper curves of the tubes even when the boiler has been filled with an alkaline solution to prevent oxidization.

[19] "The Design and Results of a 600 lb. per sq. in. Boiler Installation," by W. Nithsdale, Engineering, March 28, 1930, p. 406; also Proc. Inst. M. E., 1930.

Fig. 755, consists of a steam drum (18 in. I.D. and 3 in. thick), a mud drum, and small connecting zigzag steam-generating tubes and downcomers, and is believed to be the highest-pressure boiler relying on natural circulation. Many valuable data are expected to be derived from tests conducted with this unit.[20]

(f) The number of drums or headers required in a boiler can be reduced by using the **porcupine arrangement,** in which the tubes project outwardly from a single water-drum or set of headers in the manner indicated by the name, the outer ends of the tubes being closed. A simple arrangement of the tubes, patented in 1831 by Jacob Perkins but commonly known as *Field tubes,* is shown in Fig. 756; it consists of concentric tubes which depend vertically into the furnace from a single shell located above. The Niclausse marine boiler uses horizontal concentric tubes of this type, Fig. 757, in combination with double headers, in which one passage supplies the water to a central tube and the other passage returns the heated water to the overhead drum. One arrangement of the porcupine tubes used in the Emmett **mercury boiler** is shown in Fig. 758; to reduce the amount of mercury required, the core is made to fill the tube entirely, except for a $\frac{5}{8}$-in. central hole and a $\frac{1}{8}$-in. annular circulation space.[21] The outside tubes are $3\frac{1}{8}$ in. in diameter and $5\frac{1}{2}$ ft. long. This type of mercury boiler represents the result of many years of development and is giving satisfactory service in the Hartford plant.

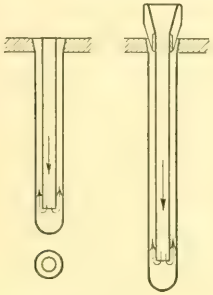

FIG. 756.—Perkins or Field Tubes.

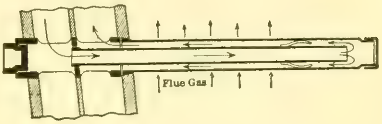

FIG. 757.—Niclausse Tubes.

(g) High rates of heat transmission and great flexibility in the rates of steam formation are secured by using **flash boilers,** in which the water is injected onto dry, heated surfaces from which it immediately flashes into steam. Although boilers of this type were extensively used by Serpollet in early steam automobiles, they do not appear to have any general field of application now. There are, however, several designs of **semi-flash boilers** which have successfully met modern requirements. Most of these boilers are of the once-through type, with long tubes in which the water passing through them flashes into steam before reaching the outlet, the rest of the tube length serving as superheater. With such boilers the water and fuel supply must be subject to continuous balanced regulation. One of the early boilers of this type is the **Belleville marine boiler,** which is shown diagrammatically in Fig. 759.

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[20] For more detailed description, see Power Plant Engineering, Dec. 1, 1931, p. 1153; or Power, Dec. 1, 1931, p. 804.

[21] For more complete description, see: "Mercury Vapor Unit," by J. Orr, Power, July 1, 1930, p. 4.

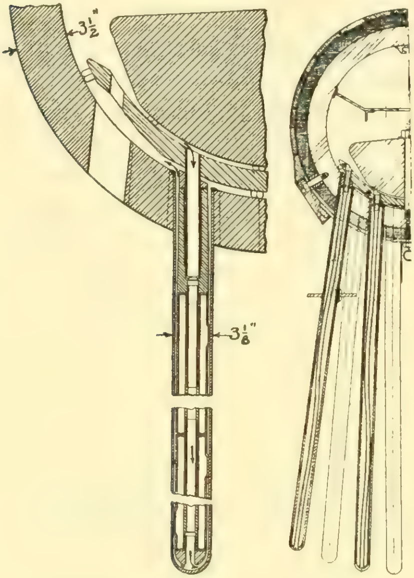

Fig. 758.—Mercury-Boiler Tubes.

(**h**) Boilers of the **film-evaporating type** are also examples of the semi-flash kind. In the **LaMont boiler** of this type the water is injected under pressure through small openings into the upper ends of long, thin, closely spaced, half-inch tubes in amounts that are small but over sufficient to maintain a film on the interior surfaces. The surplus water and the vapor formed pass from the lower end of the tubes into headers, and thence to a separating drum where the steam is liberated and delivered to the steam line. These boilers occupy but little space, and because of the high velocity and turbulence of the steam and water within them the tubes remain free from scale. This method of film evaporation can be applied also with horizontal, curved, or looped tubes. Similar construction is employed in water walls for furnaces, and in screens used to protect the shells of H.R.T. boilers from the radiation from the furnace, thus permitting such boilers to be forced to higher capacities than would otherwise be safe.[22] Small LaMont tubes can

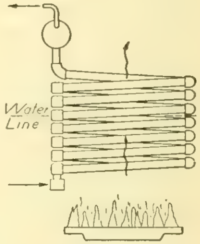

Fig. 759.—Elements of Belleville Boiler.

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[22] For more complete description, see Power, Nov. 23, 1926, p. 786.

also be placed in the larger tubes of fire-tube boilers to increase the capacities of the units.

(i) Films or thin layers of water can also be maintained in contact with heated tube walls by revolving the tubes at a suitable speed and limiting the amount of water injected into them. In the **Atmos rotating type of boiler,** as originally designed, four 12-in. tubes 10 ft. long rotated at 330 r.p.m. and into them at one end water was fed to maintain layers from $1\frac{1}{4}$ to 2 in. thick in contact with the walls, while the steam formed (at a pressure of 1422 lb. per sq. in.) occupied the central core and was discharged from the other end of the tube. The boiler was supplemented by superheater and economizer, and required very careful purification of the water used. In the later designs of these boilers, each rotating element contains a large partly filled central tube and a set of smaller tubes surrounding the former and connected thereto at their ends, and these elements revolve at a speed of about 20 r.p.m.[23]

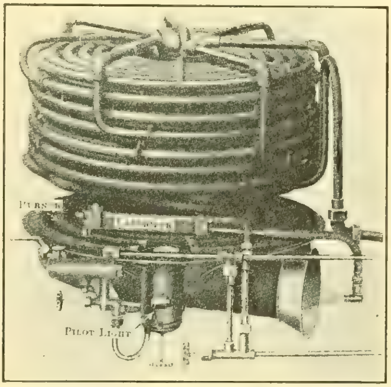

Fig. 760.—White Automobile Steam Generator.

(j) In the common arrangements of boilers the flue gases must leave the heating surface at a temperature considerably above that of the steam, with consequent loss of energy to the flue, unless supplementary heat-absorbing apparatus is used. By applying the **counter-current principle,** however, it is possible to cool the gases, and obtain the corresponding heat absorption, to any extent that is economically justifiable. The small *White automobile boiler,*[24] Fig. 760, formerly used in steam automobiles, employed this principle fully and was also of the semiflash type. These units, which were fired from below by gasoline or kerosene burners, consisted of a number of flat, horizontal tubular coils which were banked coaxially in layers that were connected in series, the boiler in effect thus consisting of one long tube. The water was pumped into the top of the boiler as fast as steam was needed; it flashed into vapor before reaching the bottom, and the steam formed issued from the lower coil in a highly superheated state. Gravitation of water from coil to coil was prevented by having upward loops formed in the connections between coils. Automatic regulators admitted more water to these boilers when the pressure fell below its normal value (550 lb. per sq. in.) and less when it exceeded this amount; thermostats controlled the fuel supply so as to maintain the temperature of the issuing steam at the desired point (750 deg. fahr.).[25]

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[23] For details of the original design, see Mechanical Engineering, April, 1923, p. 253; or Engineering, Oct. 30, 1925, p. 538. Later modifications are described in Mechanical Engineering, May, 1929, p. 379; in Power, May 28, 1929, p. 866; in "The Atmos Steam Generator," by M. Ehrlinger, International Meeting of General Mechanics, Liège, 1930, and "The Atmos Revolving Tube Boiler," by David Brownlie, Combustion, Aug., 1932, p. 30.

[24] "Steam Plant of the White Motor Car," by R. C. Carpenter, Trans. A.S.M.E., Vol. 28, 1907, p. 581.

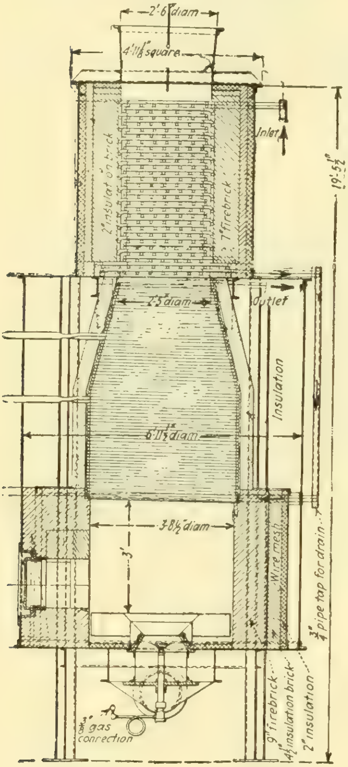

Fig. 761.—Purdue Test Unit.

Partial applications of the countercurrent principle have been made in a number of designs, as, for example, in the Parker boiler, in the heating section of the Edge Moor single-pass boiler, Fig. 740; also in the B. & W. units shown in Fig. 734 in which the steaming counterflow economizer may be considered as part of the boiler in effect, as such an economizer replaces much of the surface that would otherwise be included in the boiler elements.

(**k**) The **Purdue 1500- to 3500-lb. testing steam generator,** Fig. 761, is another example of a series, or once-through drumless unit in which forced circulation and partial counterflow are used. The feedwater enters it at the top, descends through a series of horizontal zigzag coils in the upper section, is then conducted to the bottom of the furnace wall, in the coils of which it rises and flashes into steam which becomes superheated before the outlet at the top of this coil is reached. The unit consists in effect merely of two parallel seamless tubes which are 880 ft. long and are coiled to form respectively the upper convection section and the lower radiantly heated water walls. The tubes are 1 in. O.D., $\frac{1}{4}$ in. thick, and have welded joints. Preliminary tests of this unit[26] have contributed much valuable information which was reported in the Transactions of the A.S.M.E. in 1932. Other reports are expected to follow.

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[25] For a description of the Doble counterflow automobile boiler and a proposed counterflow locomotive boiler, see Proc. N.E.L.A., 1929, p. 1222.

(1) The problems connected with handling a mixture of vapor and water which occupies a volume much larger than that of the liquid itself and from which the vapor must be separated, and those arising from the accumulation of impurities which continuously takes place in a boiler of the recirculation type, can be avoided by generating the steam at or above the critical pressure and by using for this purpose a boiler of the once-through semiflash type. The **Benson critical pressure boiler** was designed to have these features, and in addition to receive the maximum portion of its heat by direct radiation. In the early Benson arrangement, illustrated diagrammatically in Fig. 762, the boiler tubes are looped in such a manner that each tube forms a part of the furnace wall, ash screen, and roof, these elements being thus connected in series. Burners for pulverized coal, gas, or oil are located in the top of the furnace, and the secondary air enters through adjustable side openings. The furnace gases pass downward through the furnace and then ascend through annular passages containing the superheaters, reheaters, and air preheaters, and thence go to the stack above. The air for combustion cools the funnel-shaped ash pit and passes up through annular passages surrounding the unit to the air preheater, from which it is delivered to the primary and secondary air passages. As the specific volumes of the saturated liquid and vapor are equal, no drums for steam separation are necessary and hence are omitted from this unit. The structure is also free from headers, and most of the joints in it are welded. The tubes forming the looped sections are 0.8 in. I.D. and 0.256 in. thick

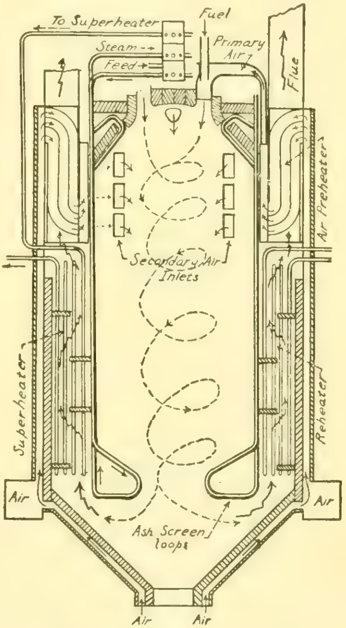

FIG. 762.—Diagram of an Early Form of Benson Boiler.

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[26] For a brief description, see Power, Dec. 1, 1931, p. 798; also "Characteristics of High-Pressure Series Steam Generator," by A. A. Potter and others, Trans. A.S.M.E., 1932. FSP–55.

and are welded into lengths of about 450 ft. The length of each superheater section is about 320 ft. The water enters the generating tubes at a pressure of 3400 lb. per sq. in. and a velocity of about 5.5 ft. per sec., and the steam is discharged from these tubes at a velocity of 57 ft. per sec. In the superheater, the entering and discharged velocities[27] are respectively 42 and 57 ft. per sec. High rates of heat transfer occur in these units because of the high density and great velocity of the working substance and because such a large proportion of the heating surface absorbs radiant energy from the furnace. The specific volume of the steam delivered is relatively very small, as the working substance at the critical state has a volume only about three times that of the entering water.

In later units the tubes in the furnace are arranged helically so as to distribute the heat more uniformly among them, and the coils are divided into two sections of which the upper one heats the water to the critical temperature and the lower one does part of the superheating; also the secondary air is admitted through the roof of the furnace, and the slag screen is omitted. The units may be located out-of-doors, as they require no enclosure to protect them.[28]

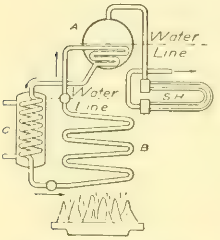

FIG. 763.—Schmidt-Hartmann Unit.

(m) A double-fluid type of boiler may be used to avoid the continuous expense of treating feedwater make-up when the raw water available is unsuitable for use in the heater circulation system. In these boilers the fluid that is heated by the furnace is sealed in a closed system of tubes, the primary system, in which it is used repeatedly without replacement; and either pure water or any other suitable liquid may be employed in this system. The vapor formed in these primary tubes is caused to pass through coils submerged in one or more vaporizing drums, or heat exchangers, in which it converts the surrounding water, the secondary liquid, into the steam that is delivered by the unit. The condensed vapor in the primary system returns to the furnace coils to repeat the cycle. Unpurified water may be used in the secondary system, as the drums and their coils can be easily cleaned of sediment and are not exposed to the furnace heat. The **Schmidt-Hartmann boiler,** the elements of which are shown in Fig. 763, is of this type and uses distilled water in the closed system. In the figure, *A* indicates the vaporizing element, *B* the heating surface of the closed circuit exposed to high temperature, *C* a feedwater heater and condensate cooler, which accelerates the circulation in the primary system, and *SH* a superheater in the secondary system. This type of steam generator has also been applied to locomotives.[29]

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[27] "High Pressure Steam Boilers," by George A. Orrok, Trans, A.S.M.E., 1928, FSP–50–32; discussion, p. 363.

[28] References: "The Benson Boiler," The Engineer, Feb. 22, 1924. "The Benson Boiler Operates Successfully," Power Plant Engineering, July 1, 1927, p. 740. "Steam Generation at Critical Condition," Power Plant Engineering, July 1, 1928, p. 718. "Tests on a Benson Boiler," Engineering, Jan. 17, 1930, p. 89. "The Benson Boiler at Langerbrugge," Mechanical Engineering, Feb., 1931, p. 146. "Benson Boiler Tested for Steamship Service," Power Plant Engineering, March 1, 1932, p. 199.

(**n**) The **Loeffler** or **steam-pumping type** of steam generator illustrates another method of producing steam from impure water without having scale form on the water-bathed surfaces that are exposed to furnace temperatures. In this process the water is evaporated solely by means of superheated steam—the furnace heat being applied only to the superheater and economizer—and a steam pump is used for circulating the vapor through the apparatus. Referring to the diagram given in Fig. 764, the saturated steam formed in the vaporizing drum is forced by means of the steam pump through the superheater coils and about two-thirds of the issuing high-temperature steam is sent back to the vaporizer, only the remaining third being delivered to the steam main. The steam pump, which is needed to overcome the resistance to flow through the apparatus and to maintain the circulation in the right direction, absorbs energy in amount dependent, among other things, on the specific volume of the steam handled; this volume in turn depends on the pressure. If the working pressure of the unit is but 280 lb. per sq. in. the pump requires for its operation more energy than the boiler can furnish, whereas it is claimed that if the pressure is 1850 lb. per sq. in.

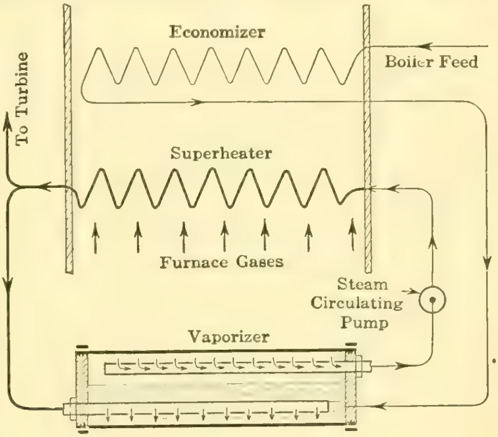

Fig. 764.—Elements of the Loeffler Unit.

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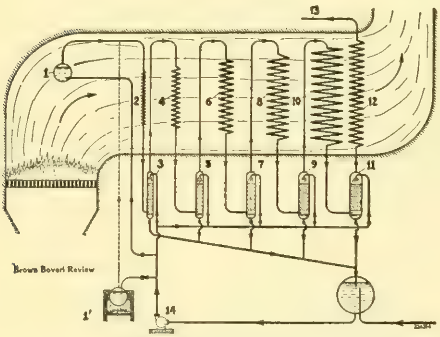
[29] References: "The Schmidt-Hartmann Boiler," by O. H. Hartmann, Power, Oct. 29, 1929, p. 696. "Schmidt High Pressure Locomotive of the German State Railway Company," by R. P. Wagner, Trans. A.S.M.E., 1929, RR–51–4, p. 29. "Loeffler vs. Schmidt-Hartmann," Power, May 6, 1930, p. 718. "Experience with High Pressure Boilers in Central Europe," Power, Oct. 21, 1930, p. 644.

only 2 per cent of the boiler output is needed for this purpose; thus it is not feasible to apply the Loeffler process except with high pressures. As the furnace gases leave the superheater at a high temperature, an economizer is also included as part of the unit. The feedwater is delivered to the one or more vaporizer cylinders which may be placed in any convenient location and are of simple, cheap construction; their heads are screwed into the ends of plain, unpierced shells which are shrunk onto them, a ring then being shrunk over the joint to counteract the stresses thus set up. The feedwater impurities are deposited only in the vaporizer cylinders, from which they are readily removed as slime. In such a unit the volume of steam circulated must be continuously regulated to conform with the changes in load and rate of firing; and, for starting, steam must be supplied from an auxiliary source.[30]

(o) The **Brown-Boveri steam generator,** or "cascade injection steam generator," the principle of which is shown in Fig. 765, also uses superheated steam to vaporize the liquid, but dispenses with the steam-circulating pump required in the Loeffler apparatus.



Brown Boveri Review

FIG. 765.—Elements of the Brown-Boveri Steam Generator.

The vaporization of the water is effected in this generator progressively in a series of elements in the following manner: A very small *exciter boiler,* 1, in the figure, is used to generate a little saturated steam at a pressure somewhat above the delivery pressure of the boiler. This steam, after being heated in the first superheater, 2, flows to the bottom of the first evaporator, 3, in which it gives up its superheat and thus converts into saturated steam some of the preheated feedwater delivered to the top of that vessel by the pump, 14, the surplus water, which is at the saturation temperature, returning to the tank, from which the pump receives its supply. The increased amount of steam thus formed passes through the second superheater, 4, goes to the second evaporator, 5, and generates therein a still larger amount of saturated steam from the water injected from the same pump—and so on through other similar elements of increasing size until the necessary amount of steam is developed and delivered in the desired state from the final superheater, 12. Instead of an excess of water being furnished to the evaporators, the necessary rate of feeding each can be controlled by a float valve, or equivalent, to keep the water level at the desired height. As the gases leave the last superheater at a high temperature, supplementary heat-absorbing apparatus should be included in the unit.[31]

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[30] References: "Steam Generation at High Pressures and Temperatures," by Loeffler, Power, March 19 and 26, 1929, p. 186 and 524; also Z.v.D.I., 1928. "Benson Boiler vs. Loeffler Boiler," by Queissner, Power, June 4, 1929, p. 936; answer by Loeffler, Power, Aug. 6, 1929, p. 219. "A 1000-hr. Trial on a 1700-lb. Loeffler Boiler," Engineering, Nov. 14, 1930, p. 632. "Data on Operation of Loeffler Boilers," Mechanical Engineering, Sept., 1931, p. 677. "The Loeffler Boiler from the Boiler Maker's Point of View," by K. Rochel, Engineering, April 24, 1931. "Steam Circulating Pump of the Loeffler Boiler," by B. Belohlavek, Power, April 5, 1932, p. 504.

Other proposed methods of generating steam which are of interest, but not yet reduced to practice, will be outlined briefly in the following paragraphs.

(p) **A combination of the Benson and Loeffler systems** into one unit has been proposed by W. Firing,[32] for the purpose of eliminating to a certain extent the disadvantages of each. The Benson system is rather sensitive to feedwater impurities, requires much energy for its feed pumps, has no thermal storage, and delivers steam at a pressure too high ordinarily for direct use, whereas the Loeffler system avoids most of these troubles, but requires a mechanical device to circulate high-pressure steam in quantities several times the actual output. In the proposed arrangement, the two systems are placed in series, with the Benson element supplying its steam to operate a jet-type of steam pump (resembling an ejector), which is substituted for the mechanical pump ordinarily used in the Benson units and in which the steam is expanded down to the pressure used in the Loeffler section. A small evaporator supplies pure feedwater to the Benson element, the remainder of the feed being delivered to the Loeffler evaporator drum.

In another combination of the two systems, proposed by N. Artsay, each system is fired separately so as to give better control.[33]

(q) The **Baumann multiple-superheater unit** uses excess superheat to generate some of the steam supplied by a unit which is planned also to maintain the issuing steam at a constant temperature under all conditions and to provide a construction well suited for generating steam at high pressure and at high temperature. The excess superheat is used to generate some of the saturated steam by effecting a heat transfer to water in an evaporator which replaces all convection water-heating surface, which otherwise would need to be provided, except that in the water-cooled furnace walls. The unit functions in the following manner: the feedwater, after first passing through an economizer, is delivered to the steam drum of the water walls, and part of it is vaporized in these walls and the rest in the vaporizer, which is connected at top and bottom with this drum. The steam from the evaporator and drum passes through a primary superheater wherein it is raised to a temperature above the final delivery value; it is then passed through the tubes in the evaporator which desuperheats it somewhat; and, finally, it flows through a second superheater which raises its temperature to the desired final value, which is thermostatically regulated by raising or lowering the water level in the evaporator. The design of the apparatus has many ingenious and attractive features.[34]

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[31] References: "Developments in Foreign Boiler Design," Power Plant Engineering, Nov. 15, 1929, p. 1217. "Brown-Boveri High-Pressure Boilers," Proc. N.E.L.A., 1929, p. 1244. "The Brown-Boveri Cascade Injection Steam Generator," Brown-Boveri Review, Jan., 1928, p. 34; Jan., 1931, p. 64. "High Pressure Boilers in Europe," Power Plant Engineering, April 1, 1932, p. 289. "Instalment Steam," Power Plant Engineering, Aug. 1, 1932, p. 602.

[32] Power, April 15, 1930, p. 587; adverse discussion, Power, July 8, 1930, p. 63.

[33] "More on Benson and Loeffler," by N. Artsay, Power, May 6, 1930, p. 714.

(r) **Electric boilers** may be used to generate steam economically under special conditions. These boilers are built in three types: (1) immersion, (2) tubular, and (3) electrode. In the first type the heating unit is made of resistance material, is immersed directly in the water, and uses either alternating or direct low-voltage current; these units are built in small sizes, commonly under 200 kw. For slightly greater capacities, the second type is used; in these the resistance heating units are kept dry and are generally located within the tubes of horizontal return tubular boilers. For still larger capacities and higher voltages, the third type is preferred; in this type, electrodes are inserted in the water, which therefore becomes the electrical resistance. Alternating current must be used in the electrode boiler in order to prevent the excessive formation of explosive hydrogen and oxygen mixtures due to electrolysis; and unless this type of boiler is bled, the concentration of salts in solution will increase and the resistance decrease. An automatic system of controlling the electrical resistance of the water, and hence the steam pressure, has been developed.[35] The average daily efficiency of the electrode boiler with constant pressure control is from 90 per cent at 25 per cent capacity to 98 per cent at maximum capacity.

The economic use of the electric boiler is confined to regions near extensive hydroelectric developments where the price of coal may be high, as in certain paper, pulp, and electrochemical plants near the St. Lawrence River. Hydroelectric plants may economically use off-peak or surplus water power in the electric generation of steam; also, consumers of large blocks of electrical energy purchased on a flat rate may find it advantageous to generate steam electrically.[36]

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[34] For a description of this proposed steam-generator and a very valuable discussion of the future possibilities in utilizing steam with high pressures and high temperatures, see "Future Development of the Steam Cycle," by K. Baumann, Proc. Inst. M.E., Dec., 1930, p. 131; or the reviews of this paper in Engineering, Nov. 21, 1930, p. 664, and in Power, Dec. 23, 1930, p. 1019.

[35] "Electric Steam Generators Controlled Automatically," Power, Sept. 29, 1931, p. 458.

(s) **Steam accumulators,** or **heat accumulators,** may be used in conjunction with steam boilers to store energy during periods when the output of the boilers exceeds the demand, and to restore or supply energy when the reverse conditions occur. The modern forms of these accumulators use water as the medium for storing heat, and may be divided into two general groups, *variable-pressure* and *constant-pressure* accumulators.

In general, the **variable-pressure accumulator** is a receiver, containing a large volume of water, connected between two steam circuits of different pressure. When charging, the accumulator is supplied with high-pressure steam; when discharging, it gives up steam to the low-pressure circuit. The operation of the accumulator is based upon the principle that saturated water will partly evaporate, or supply steam, if the pressure be reduced; hence the temperature of the remaining water will decrease. The cooled water may then absorb heat if the pressure is increased, i.e., it can condense some high-pressure steam. Where any boiler is operating under large fluctuations in pressure, it is, to some extent, a heat accumulator.

For example, if the pressure on saturated water in a boiler be decreased from 415 to 325 lb. per sq. in. abs., 1.7 lb. of steam will be obtained from each cubic foot of water. The variable-pressure heat accumulator might be arranged to work between pressures of 165 and 75 lb. per sq. in. abs.; for this case 3.7 lb. of steam would be obtained from each cubic foot of saturated water.

The storage capacity of an accumulator is greater than that of a boiler, but the pressure of the accumulator is lower, and a large pressure fluctuation would be undesirable in a boiler; hence heat storage capacity is more economically supplied by an accumlator than by a boiler. The variable-pressure accumulator may be used in central power plants to assist in carrying sharp peak loads of short duration; the boilers of the plant can then be of smaller capacity and can operate at more nearly constant load and with better efficiencies.[37]

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[36] References: Power, Feb. 6, 1923, p. 208; July 8, 1930, p. 62; Sept. 30, 1930, p. 544; May 12, 1931, p. 728. Mechanical Engineering, June, 1927, p. 692. Engineering, Oct. 10, 1930, p. 460. Power Plant Engineering, June 1, 1930, p. 623. Combustion, Aug., 1930, p. 33; Sept., 1931, p. 53. N.E.L.A. Special Report of the Industrial Heating Committee, March, 1930.

[37] "The Peak-Load Problems in Steam Power Stations," by A. G. Christie, Mechanical Engineering, Vol. 50, 1928, p. 914.

The Ruths accumulator is of the variable-pressure type and has many industrial applications where steam is being generated at high pressure for use in non-condensing or bleeder turbines, the exhaust steam from which is used in process work. The rapid fluctuations in steam demand may be smoothed out and the unequal requirements for power and process steam may be balanced without waste. The accumulator is a cylindrical tank of large volume with hemispherical ends, and needs no building protection; it is generally installed out-of-doors because of its great size.

A typical application[38] of an accumulator to a pulp and paper mill is shown in Fig. 766. The digestors use steam at a pressure of 150 lb. per sq. in. gage, and their demands fluctuate very rapidly. The paper machines use steam at a pressure of 10 lb. per sq. in. gage, and their steam demand is quite steady. The accumulator works between gage pressures of 150 and 60 lb. per sq. in. Oil-actuated relay valves govern the operation of the accumulator; the high-pressure turbine exhausts into the accumulator when the demands of the digestors are light, and the accumulator furnishes steam at a steady pressure to the paper machines to meet their requirements. The boilers can operate at a capacity, nearly constant, that meets the average steam requirements of the mill. In this illustration, if 25,000 lb. of steam storage capacity were required, and a storage of 3.7 lb. of steam per cu. ft. of water exist between these pressure limits, the volume of water required would be 6750 cu. ft., and the tanks might be made about 14 ft. in diameter by 50 ft. long. The steam line inside the accumulator connects with several charging nozzles and circulation pipes to give a uniform and noiseless heating of the water.

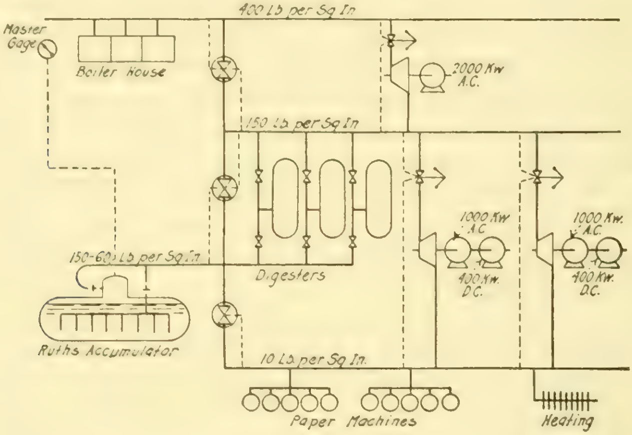

FIG. 766.—Application of Accumulator to a Pulp and Paper Mill.

Constant-pressure accumulators store heat in water during the charging period, but *deliver water*, not steam, during the discharge period, the pressure in the accumulator remaining substantially constant. In the Kiesselbach constant-pressure system, the accumulator is connected to the boiler by a circulating and an overflow pipe, through which water is continuously circulated by a pump. A pressure-equalizing pipe connects the accumulator and the boiler. Separate feed pumps supply feedwater to the boiler. If there is a decrease in the demand for steam from the boiler, the quantity of cold feedwater supplied to the boiler is increased, the water level in the boiler rises, and some heated water flows into the accumulator. As the demand for steam increases, the quantity of cold feedwater supplied to the boiler is cut down, the circulating pump supplies saturated water at the boiler pressure from the accumulator, and only the latent heat of vaporization need be supplied by the boiler to change this into steam. The Kiesselbach accumulator is, therefore, a *feedwater accumulator*, and it has proved satisfactory in increasing the output and improving the efficiency of existing plants.

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[38] "The Ruths Steam Accumulator," by R. A. Langworthy, Trans. A.S.M.E., 1927–28, FSP–50–33, p. 68.

The decision as to whether an accumulator should be installed in any given plant depends on the variations in the load and the cost of fuel.[39]

(t) The **merits** claimed for many of the newer designs of steam-generating units already in operation, and for those proposed, are still matters of some controversy. The actual performance of these units under service conditions will be watched with interest and will eventually determine the conditions for which each design is best suited. The analysis of these designs and of their operating characteristics may suggest other forms of apparatus of still greater value.

481. The Water Circulation System.—(a) Difficulty is seldom encountered with the circulation of the water within boilers or with the stability of their water level when they are operated uniformly and at moderate rates. But with high forcing and rapidly changing demands, the circulation may be erratic and inadequate, and the water level may fluctuate widely, sometimes dropping out of sight in the gage glass or becoming so high that much moisture is carried out with the steam delivered. All parts of the heat-transmitting surfaces should always be protected by adequate circulation over them of the heat-absorbing fluid (liquid or vapor, as the case may be), regardless of the conditions of operation; and all boilers should be designed and operated with this in view. Each arrangement of unit, however, has its own special circulation and water-level problems. The discussion here will be with respect to the more usual types of units.

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[39] References: The Fuel Economist, Dec., 1928. Power Plant Engineering, July 1, 1930, p. 734. Power, Nov. 25, 1930, p. 839; March 10, 1931, p. 394. The Brown-Boveri Review, Dec., 1931, p. 347. Trans. A.S.M.E., FSP-50-33, 1928, p. 68; FSP-51-24, 1929, p. 109.

(b) In general, **adequate circulation** in a steam-generating unit is of prime importance because it affects directly the (1) *safety,* (2) *reliability,* (3) *steaming capacity,* (4) *efficiency,* (5) *size,* and (6) *cost* of the unit. Deficient circulation may cause the plates and tubes exposed to the fire and hot gases to become overheated and, as a result, lose strength, blister, bulge, or burn, and possibly rupture, especially when the unit is forced to high capacity. Poor circulation also reduces the heat transmission rate; this causes the exit gases to leave at high temperature, and thus reduces the capacity and efficiency of the unit and necessitates installing larger and more costly apparatus for obtaining the same output. It is particularly important that the velocity of circulation be sufficient to prevent the adherence to the wall surfaces of blanketing bubbles or films of steam, as these have little conductivity. If the surfaces are kept free from scale and steam films, the temperature of the outer surface of the wall metal will not be much above that of the contiguous water within, and the apparatus can be driven safely to very high capacities. In general, the more rapid the circulation the more effective is the scouring action, with the result that the wall temperature is lower and the rate of heat absorption greater.

(c) Although it would appear that *overheating* is more prone to occur in water walls or boiler parts that are not only swept by the hottest gases but are also exposed to direct radiation, yet, under intensive operation, this trouble may occur instead in less exposed parts if there are localized deficiencies in the circulation, such as those due to either *inadequate passages* or *short-circuiting.* For example, when banks of tubes are connected horizontally to common headers having small risers, short-circuiting may occur through some of the upper tubes under high forcing, with the result that there may be little or no flow through the middle tubes, which may consequently become steam bound and overheated.[40] Trouble with circulation may arise not only from forcing a boiler beyond the capacity for which its circulation system was intended, but also from operation under other than the normal conditions. Thus, a 1400-lb. unit may have faulty circulation when operated at 300 lb. per sq. in. pressure, because the volume of the steam entrained in the water in the risers is very much greater at the reduced pressure.

(d) The **mechanism of natural circulation,** or the action by which positive flow is established and maintained in a boiler, is due to the difference in the average densities of the ascending and descending columns of fluid. This difference in densities, for boilers operating under pressures less than the critical, is *primarily* due to the presence of a large number of *vapor bubbles* in the ascending column and relatively few in the descending one; *secondarily*, the difference is due to the *higher temperature of the liquid* in the ascending column relative to that in the descending one.

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[40] Bureau of Mines Bul. 214, Tests of Marine Boilers, p. 166.

For example, consider a column of fluid 10 ft. high that is partly steam and partly water uniformly mixed and at a common temperature of 382 deg. fahr., which is approximately the saturation temperature for a pressure of 200 lb. per sq. in. abs.; also consider a column of water of the same height and at a temperature of 300 deg. fahr. connected to the first column to form a circuit. Then by the aid of the steam table the following results may be calculated for various percentages of steam:

Per cent, by volume, of vapor in ascending column	0	10	50
Static pressure difference available to produce circulation, pounds per square inch	0.21	0.59	2.08

Clearly, in this example, by having 50 per cent of the volume of the first column occupied by the bubbles, the static pressure difference is ten times greater than when this column has no vapor in it. (In actual boilers the differences in temperature of the two columns depends on the type and rate of operation.)

The preceding values serve to show how rapid circulation may be produced by intensive application of heat to the base of an ascending column of fluid in a boiler. On the other hand, there is danger of burning a boiler tube when the percentage of vapor in it becomes too large, especially if the bubbles are not small and well diffused. Furthermore, with a large percentage of vapor in the tube, the velocity of the mixture becomes so high that the resistance to flow soon equals the available force producing circulation. Consequently, with intensive application of heat near the bottom of the ascending column, steam may be formed at such a high rate that part of it may even pass back through the descending column of water in the form of large bubbles. The boiler is then sometimes called *steam bound*, and a tube is likely to be burned.

In general, the most satisfactory circulation is obtained when the bubbles are small and well diffused throughout the ascending column. The whole subject of circulation in boilers and water walls is complicated because it involves the resistance to flow of a two-phase fluid through various types of circuits, the transmission of heat from a gas through a metal to this fluid, and the variable temperatures, velocities, and densities throughout the circuit. Actual test data on natural circulation in boilers and water walls are therefore very meager, but the references given at the end of this section will be useful in studying the subject.

(**e**) The **fluctuations in the water level** in water-tube boilers under various conditions can be studied in connection with the simple element shown in Fig. 767. In this figure, 1–1′ represents the level of the water when it is cold and quiescent, and 2–2′ is the level when the liquid has been heated uniformly throughout to a higher temperature, say just to the boiling point. If circulation is now established in a clockwise direction the liquid then becomes inclined, as shown by line 3–3′, and the *apparent water level,* as indicated by the gage, will be at 3. With the formation of steam and more rapid circulation the apparent water level will be at 4. Thus, although the amount of water in the apparatus has not been changed, the gage glass has shown four different water levels. Fluctuations in level may also take place, owing to variations between the rates of steam withdrawal, firing, and water feeding, and to the chilling effect of the entering water on the mass already in the apparatus. Erratic changes in water level may be brought about not only by improper boiler feeding by hand control, but also by the faulty functioning of automatic feed regulators of the type operated solely by fluctuations in the water level, when such regulators are used on units fired by oil, gas, or pulverized coal, and are subject to sudden wide changes in demand. It is preferable that the water be fed continuously, or frequently and in small quantities, at rates which accord always with rates of steam withdrawal.

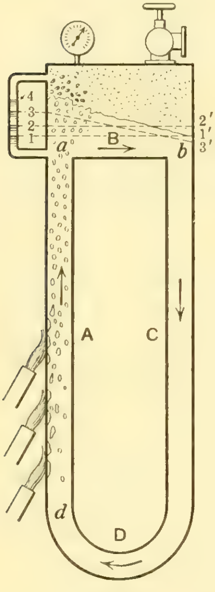

Fig. 767.

The difference in the water levels in the various parts of boilers alters with changes in the rates of forcing and has an important influence on the steam space left in the drums, on the areas of the liberating surfaces (when the walls are curved), and on the dryness of the steam delivered. The differences in water levels are indicated in Figs. 739, 768, and 769.

(f) The **formation of steam** in riser *A* in Fig. 767 takes place in the following manner: If the water arriving at *d*, at the base of the riser, is at the saturation temperature corresponding to the pressure at that level, then even without further application of heat some steam will form as the water ascends, this being due to the decrease in the static head. The steam generated by the application of heat to the riser forms on the surface of the latter, and, unless constantly scoured away by the water circulation, it adheres thereto until the bubbles formed become large enough to break away of their own accord. As these steam bubbles ascend in the riser, they expand slightly in volume, owing to the decreasing static head, and give up some of their energy, which vaporizes other particles of the water. When the vapor in the drum is under pressure, the pressure existing at any level is the sum of the drum pressure and the static head; and the vaporizing temperature at that level is the one corresponding to the combined pressure.

(g) When the **steam formation becomes very rapid** the discharge from riser *A* may become so vigorous as to cause **spouting** to occur through the body of the liquid immediately above, thus further disturbing the water level and also causing **priming**, i.e., the carrying of moisture into the steam space. Spouting and priming may be avoided by providing disengaging or **liberating surfaces** of adequate area and by baffling the steam discharged in such a manner as to prevent the localization of its release. Moisture in the steam can also be caused by **foaming** due to the presence of impurities in the water. To remove moisture from the steam before it is discharged from steam drums, they are generally provided with dry pipes, steam baffles, or steam separators.[41]

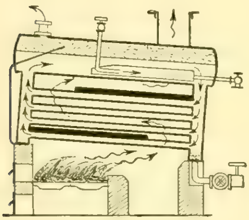

FIG. 768.

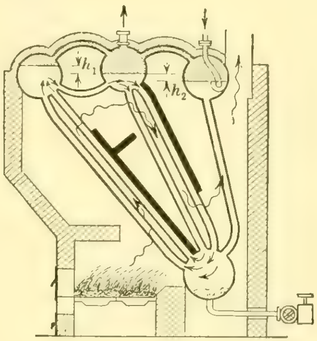

FIG. 769.

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[41] References on Circulation: "Circ. in Horizontal Water-Tube Boilers," by Bancel, Trans. A.S.M.E., Vol. 37, p. 1011. "Heat Transmission through Tubes," by Croft, Bul. 168, Eng. Exp. Sta. U. of Illinois (1927). "Effect of Radiant Heat on Water Cooled Furnace Walls," by Christie, Power, May 31, 1927, p. 841. "Water Screen Test, Cahokia Station," Proc. N.E.L.A., 1925, p. 1510. "Tests of Marine Boilers," Bur. Mines Bul. 214. "Velocity of Circulation in Water Wall Tubes," by Smail, Power, March 5, 1929, p. 411. "Circulation in Water-Cooled Furnace Wall Circuits," by Orrok and Artsay, Proc. N.E.L.A., 1929, p. 1265. "High Pressure Boilers," by Orrok, Trans. A.S.M.E., 1928, FSP-50-32, p. 47. "Correct Boiler Design," by K. Toensfeldt, Combustion, Sept., 1929, p. 40. "Foaming and Priming," by C. W. Foulk, Industrial and Engineering Chemistry, Sept., 1929; also Annual Meeting A.S.M.E., 1931. "The Basic Laws and Data of Heat Transmission (VI—Evaporation and Condensation)" by W. J. King, Mechanical Engineering, Aug., 1932, p. 560.

482. Boiler Heating Surface.[42]—(a) When boiler surface is operated at the so-called manufacturer's rating, the *average* rate of heat absorption for the total surface is 3350 B.t.u. per hr. per sq. ft. (see p. 441), a rate which is met in ordinary boilers when operated moderately. The most economic rate with simple boilers is slightly higher. Central station boilers which are supplemented by other heat-absorbing apparatus are proportioned to have average absorption rates of several times this value (e.g., from $1\frac{1}{2}$ to 5) under usual loading; and when meeting peak or emergency demands they are forced to still higher rates. The extent of surface for a boiler of desired steam-generating capacity can be determined most simply by finding the rate at which the total amount of heat must be supplied, and dividing this by an assumed average transmission rate per square foot which has been found by experience to be suitable for the conditions that are to prevail. Such method is applicable when using standardized boilers under the ordinary conditions. In other cases, especially when it is necessary to provide for intensive operation and the greatest economy in performance and investment, the determination of the extent of surface should be based on a careful analysis of all factors involved.

(b) **A detailed analysis** of the boiler heating surface may be conveniently made by considering this surface as an aggregation made up of successive layers of tubes, or of elements of surface, each transverse to the direction of gas flow, and all arranged in series in the order of decreasing effectiveness; also by considering the total heat absorbed by the boiler to be the summation of the amounts absorbed by these elements. The part of the surface exposed to the furnace, i.e., that which " sees " the interior thereof, transmits heat at the highest rate of any part, this heat being received mostly by direct radiation from the fuel and furnace gases, and by radiation and reflection from the furnace walls; although this surface also receives a small amount of heat by convection, it is commonly designated as the **radiant surface** of the boiler. The remaining boiler surface receives heat mainly by convection from the furnace gases and hence is known as the **convection**

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[42] The definition of boiler heating surface, and instructions for the measurement of this surface, as given by the A.S.M.E. Code on Definitions and Value (1930), are quoted on p. 436. As the measurement of this surface is always made on the less-effective or gas side of the walls, it is made on the outside area of water tubes and on the inside area of fire tubes. Boiler tube sizes are designated, however, according to external diameters, hence commercial computations of tube surface are commonly based on these diameters. In the case of return tubular boilers the effective heating surface is therefore less than that thus computed, and should be the one used when making analyses involving heat transfer through such surfaces, rather than the commercial area.

surface of the boiler, although it also receives some heat by radiation (see example 3, p. 281).

(**c**) **Heat transfer by radiation** from solid masses, gases, vapors, and suspended matter was discussed in Sect. **337**, and the principles there given can be applied in analyzing radiant heat transmission to the boiler surface in much the same manner as was done when the water walls of furnaces were under consideration (Sects. **467, 468,** and **469**). When the furnace has water walls which are connected into the boiler circulation, the fraction of the furnace heat absorbed by the furnace envelope, including the radiant boiler surface, is indicated by the values of u obtained from the curves previously given in Figs. 697 and 698 for the types of furnaces to which they apply. When the furnace is not water cooled, the value of u corresponding to $\psi = \frac{1}{6}$ is the fraction of heat absorbed by radiation by the boiler tubes forming the furnace roof. The rest of the heat liberated in the furnace and not lost to the outside is available for transmission by the gases to the convection surfaces of the boiler and other apparatus which may be appended thereto.[43]

(**d**) The **transfer of heat by convection** to boiler walls is illustrated in Fig. 770. At (A) in this figure[44] are shown, quantitatively, typical temperature gradients through the stagnant gas film, the tube wall, and the stagnant water film, all measured at a point near the beginning of the convection surface; and from these results it will be observed that the respective temperature drops through the gas film, tube, and water film are about as 52 : 1 : 0.37. This shows that nearly all the resistance to heat transmission from the gas to the water is encountered in the gas film; hence, when the plate is reasonably clean, the resistance offered by it and the scale can be neglected in ordinary analyses. Corresponding gradients near the boiler outlet are given at (C) in the figure; and at (B) are shown the variations in the gas temperature and in the temperature difference (θ) throughout the length of the convection surface. The effectiveness of the different unit areas of the surface are proportional to the values of the mean temperature difference existing between the two fluids for any given length of gas travel.

The foregoing is for a boiler which is operated at a moderate rate with working pressure of 125 lb. per sq. in. gage, and the highest temperature of the external surface of the tube metal is less than 400 deg. fahr., which is a safe value. As an example of an extreme case,[45] consider the temperature drops through a 3-in. water-wall tube which is 0.35 in. thick and is assumed to transmit heat at the rate of 75,000 B.t.u. per hr. per sq. ft. to water which is at a pressure of 1400 lb. per sq. in. and flows through the tube at a velocity of 8 ft. per sec. The estimated drops are as follows: through the metal, 84 deg. fahr.; through scale $\frac{1}{32}$ in. thick, 195 deg. fahr.; from scale to water, 53 deg. fahr.; total drop, 332 deg. fahr. As the saturation temperature at 1400 lb. per sq. in. is 587 deg. fahr., the temperature of the outer surface of a tube is 919 deg. fahr., which would eventually lead to failure if the tube is of the usual material. To make the conditions safe in this case, either the formation of scale must be avoided, a material less affected by temperature must be used, tube protection must be provided, or smaller and thinner tubes must be substituted.

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[43] For a discussion of transmission of radiant heat to boiler tubes, as well as to furnace water walls, see "Effect of Radiant Heat on Water-Cooled Furnace Walls," by A. G. Christie, Power, May 31, 1927, p. 841.

[44] Based on data and illustrations contained in Bureau of Mines Technical Paper 114 (1915) on "Heat Transmission through Boiler Tubes," and that given on p. 438 of Trans. A.S.M.E., 1916. In (A) and (C) of the figure the horizontal distances have no significance as the film thicknesses are shown greatly exaggerated.

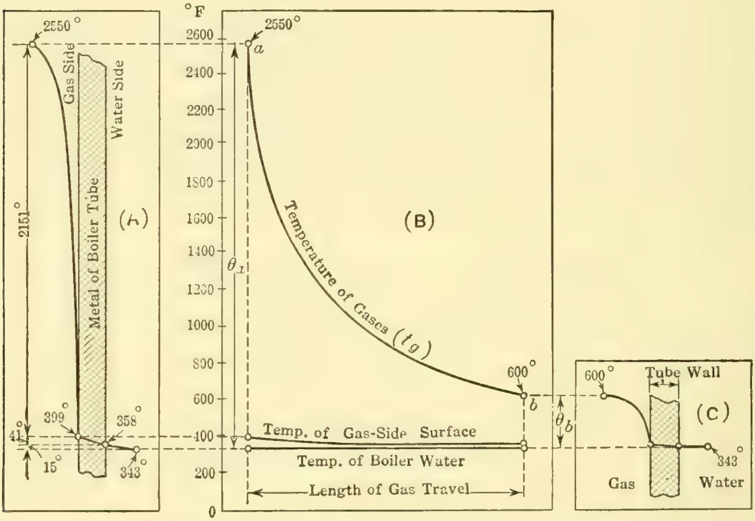

FIG. 770.—Illustrative Temperature Gradients.

(e) **Computations of heat transmission by convection** in boilers may be based upon the treatment given in Sects. **334–336, 338–341.** A careful analysis requires consideration of the dimensions, shape, and positions of the surfaces; the gas temperatures, velocity, density, viscosity, and specific heat; various thermal conductivities, and the temperature differences between the gases and the metal surfaces. For ordinary purposes, however, the computations may be based upon the equation:

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[45] From "Operating Experiences with 1300-lb. Steam Pressures," by J. Anderson, Engineering, Jan., 1928, pp. 25, 55; also Proc. N.E.L.A., 1928, p. 1168.

$$UA\theta_m = w_h c_h(t_a - t_b) \quad . \quad . \quad . \quad . \quad . \quad . \quad (583)$$

obtained by equating Eq. (431) to Eq. (432), p. 288. This method makes use of the overall coefficient (U), which takes care of many of the factors about which there is uncertainty, and which therefore should be based on reliable data obtained from similar boilers functioning under similar conditions. The notation and units used with this equation are the same as those given on p. 288, t_a being the temperature of the gas entering the convection surface, and t_b that of the gas leaving. With simple boilers t_b is usually 30 to 100 deg. fahr. above the saturation temperature of the steam when the operation is moderate, and 200 to 300 deg. above when highly forced.[46]

For most applications the values of U established by the experiments conducted by Jordan and by the Babcock & Wilcox Company (and previously mentioned on p. 285) are satisfactory. The empirical equation representing the results may be rewritten in the following form:

$$U = C + D(w/A) \quad . \quad . \quad . \quad . \quad . \quad . \quad (584)$$

in which

w = weight of gas flowing, in pounds per hour.

A = average cross-sectional area of gas passage through the boiler, in square feet.

C = a constant ranging from 1 to 3.5. Its magnitude increases as the tube diameter decreases and is greater for water tubes than for fire tubes of the same size. In water-tube boilers it becomes greater as the space between tubes is decreased.

D = a constant ranging from 0.0005 to 0.004. It is greater for water tubes than for fire tubes, and it becomes larger with increase in tube diameter and in temperature difference between the gas and water.

For standard types of longitudinal-drum water-tube boilers, C is about 2 and D is about 0.0014 when the average rate of heat transfer is from 3350 to 5000 B.t.u. per sq. ft. per hr.

Curves indicating the way the gas temperature decreases with the percentage of heating surface traversed in ordinary boilers, are given in Fig. 771, due to Coutant.

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[46] For data, see Power, Nov. 6, 1928, p. 752.

(f) For **circular tubular gas passages** of radius r it can be shown[47] that the value of θ, in terms of θ_a and the distance x passed over, is given approximately by an equation of the form

$$\theta = \theta_a \epsilon^{-\frac{cx}{r}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (585)$$

in which c is a constant and $\epsilon = 2.718$. Since the heat available for transmission by convection is proportional to θ_a, the efficiency of the convection surface of length l is, approximately,

$$e_c = \frac{\theta_a - \theta_b}{\theta_a} = 1 - \epsilon^{-\frac{cl}{r}} \quad . \quad . \quad . \quad . \quad . \quad (586a)$$

$$= 1 - cr/l \text{ (closely).} \quad . \quad . \quad . \quad . \quad . \quad . \quad (586b)$$

These equations show that flues of small diameter absorb more heat per pound of gas passing through them than tubes of large diameter and of the same length. In the smaller tubes the gas particles are, on the average, closer to the wall and therefore make more contacts with the surface in a unit of time than they make in a larger tube; also the higher velocity in the small tube increases the scrubbing action which reduces the thickness of the gas film. Accordingly, in a multitubular boiler it is preferable to use, within reasonable limits, many tubes of small diameter and of considerable length, rather than the reverse; and in water-tube boilers to have comparatively small cross-sections for the gas passages between the tubes and long paths for the gas to travel. However, although the use of high gas velocities reduces the amount and cost of the boiler heating surface required for a given output, the gain thus effected may be wholly offset by the cost

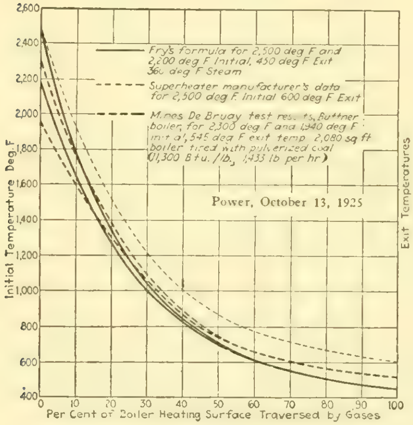

Fig. 771.—Comparison of Initial and Exit Temperatures Determined by Formula and Tests.

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[47] For the derivation of this equation and more extensive discussion, see Bureau of Mines Bul. 18, "The Transmission of Heat into Boilers" (1912), by Kreisinger and Ray; or Perry's "The Steam Engine," p. 588.

of producing the high gas velocities if the latter are made too great; thus there are economic limits beyond which it does not pay to go in proportioning the passages. The best combination of passage area and length for ordinary boilers has been established by wide experience. The heat-absorbing area needed is also reduced by so arranging the gas passages that none of the surface is ineffective, and that the baffles will remain tight.

The *average rate* of heat absorption in boilers installed in the United States in 1928–29 ranged from 2500 to 19,000 B.t.u. per hr. per sq. ft.; the maximum rate varied from about 7000 to as much as 26,000 B.t.u. per hr. per sq. ft.[48]

(g) The **most economical extent of boiler surface** can be determined by the method indicated in Fig. 772, in which the curve *a* shows how the increase in boiler surface affects the annual expense for the fuel consumed (and also for the power required to operate the boiler auxiliaries, for labor, maintenance, repairs, and supplies, if these quantities vary with size of unit); curve *b* indicates how the annual fixed charges (or carrying charges to cover interest, depreciation, taxes, and insurance based upon capital invested in boiler) increase with the surface; and curve *c* is the sum of *a* and *b* —thus giving the total annual cost for these items with respect to the amount of the surface used. The shape and position of curve *c* depend upon the annual output of the boiler, character of the loading, unit costs of fuel and boiler surface, and the efficiency of the boiler and furnace. The economic surface is determined by the abscissa of lowest point on curve *c*; and this surface is somewhat less than that giving the best efficiency. If the fuel is expensive, more surface should be used than if it is cheap; and if the energy in the gases at the lower temperatures can be used more economically in other apparatus than in the boiler, the surface of the latter should be reduced and that of the other apparatus should be substituted for it.

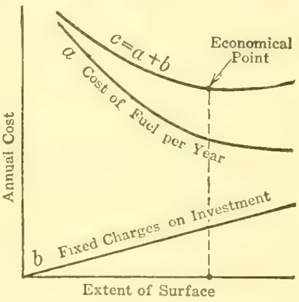

Fig. 772.

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[48] Proc. N.E.L.A., 1930, p. 1066. Other references: "Heat Transmission in Boilers," by Royd, Constable & Co. "On the Transmission of Heat in Boilers," by Hedrick and Fessenden, Trans. A.S.M.E., 1916, p. 407. "Heat Transmission through Boiler Tubes," by Croft, U. of Ill. Exp. Sta. Bul. 168 (1927). "The Transfer of Heat between a Flowing Gas and a Containing Fluid," by Fry, Trans. A.S.M.E., 1917, p. 709. "Heat Transfer in Locomotive Superheater," by Fry, Trans. A.S.M.E., 1930, RR–52–3, p. 13. "Heat Transmission," by King, Mech. Eng., Aug., 1932.

(h) When a **boiler and economizer** are to be used in combination, the **economic extent of surface** of each can be found by the method[49] shown in Fig. 773. In this figure the ordinates of the points shown on the small curve at the left give the respective *annual carrying and operating costs* for the combination of a boiler of size *A* with each of five different extents of economizer surface. The other small curves are similar ones for boilers *B*, *C*, *D*, and *E*, of progressively smaller sizes, each with various extents of economizer surface; the low point on each of these curves determines the best amount of economizer surface to use with the boiler for which the curve is drawn. Passing a line through these low points of the small curves results in the final curve, the low point of which determines the best combination of both economizer surface and boiler surface, the latter being obtained by interpolation. If the boiler surface is followed by an air preheater, instead of by an economizer, the procedure would be the same.

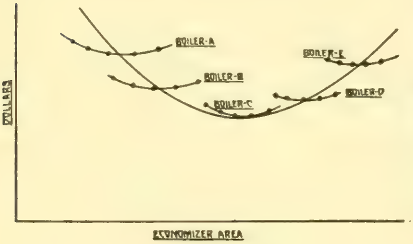

FIG. 773.

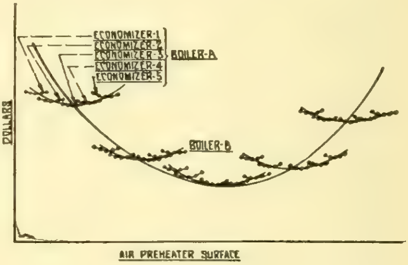

FIG. 774.

(i) The **proper distribution of heating surface** in a unit containing a *boiler*, *economizer*, and *air heater* can be determined by extending the method just considered. In Fig. 774, the small curve at the left is for the combination of boiler *A* and economizer No. 1 with various extents of air heater surface, the latter being the abscissas. The adjacent small curves are for the combination of boiler *A* and other economizer sizes, each with various amounts of air preheater surface. The other families of curves are found in a similar manner for boilers of smaller sizes, and with various extents of economizer and air preheater surface. The low point of the locus drawn through the respective low points of the groups of curves for the various boilers determines the economic extent of boiler, economizer, and air preheater surfaces.

Viewed in another way, each unit area of heating surface added at the outlet of a boiler, economizer, or air preheater is less effective than an equal amount of similar surface ahead of it, but costs just as much; hence the economic limit of surface in each apparatus can also be determined by finding the extent beyond which the heat saved annually by additional area will not justify the accompanying increase in cost; or the economic limit is that beyond which the heat saved by extra area can be absorbed as effectively by another kind of surface that costs less.[50]

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[49] "Engineering Analysis as Applied to the Selection of Type and Size of Power-Plant Equipment," by J. N. Landis, Trans. A.S.M.E., 1928, FSP-50-42, p. 193.

The distribution of heating surface also depends to a large extent on the *pressure* and *temperature* at which the steam is generated. Increasing these factors has the following effects: (1) The extent of evaporating surface becomes relatively less, as shown in Fig. 775, due to the lower latent heat at high pressures; (2) the extra recovery apparatus assumes greater importance because the products of combustion leave the boiler and superheater surface at a higher temperature; (3) the economizer becomes less able to reduce the flue gases to the low temperature desired because the regeneratively heated water comes to it at a higher temperature; and (4) the air preheater is called upon to absorb a greater proportion of the heat generated in the furnace. Fig. 775 should be interpreted more as indicating the trend rather than as representing absolute values.[51]

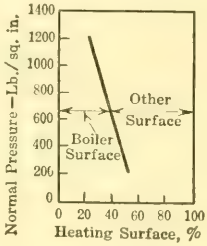

FIG. 775.—Boiler H. S. as a Percentage of the Total Surface.

The distribution of surface, in per cent of the total, and the relative costs (both total and per square foot) of the different elements as compared with that of the single pass boiler taken as 100, are given in Table LXVII for a unit resembling that in Fig. 539 designed for a steam pressure

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[50] For more detailed discussion of this and other methods employed, consult the following references: "The Selection of Steam Boilers," by W. A. Shoudy, Combustion, Nov., 1927, p. 285. "Economical Combination of Boiler, Economizer, and Air Preheater," by G. Wilkes, Southern Power Journal, Dec., 1930, p. 56. "Economizers and Air Heaters," by O. Kubalck, Combustion, Sept., 1928, p. 147. "Dividing Heating Surface among Boiler, Economizer and Air Heater," by L. Helander, Power, March 4, 1930, p. 358. "Factors that Influence the Choice of a Boiler," by J. B. Crane, Power, Nov. 6, 1928, p. 750. "The Economics of Air-Preheater Applications," by F. M. Van Deventer, Trans. A.S.M.E., 1927–28, FSP–50–16, p. 123. "Economics of Heat Transfer," by W. J. Wohlenberg, Trans. A.S.M.E., 1930, FSP–52–41, p. 351. "Some Economic Considerations of Water-Wall Installations," by Ollison Craig, Trans. A.S.M.E., 1930, FSP–52–27, p. 185. "Effect of Large Boilers at High Capacities on Operating Characteristics and Investment in Boiler Plants," by Frank S. Clark, Trans. A.S.M.E., Sept.-Dec., 1930, FSP–52–31, p. 243. "Conversion of Coal-Fired Boilers and Furnaces to Gas Firing," by William D. Edwards, Trans. A.S.M.E., Jan., 1932, FSP–54–3, p. 23. "Comparative Performance of a Large Boiler Using Oil and Natural-Gas Fuels," by F. G. Philo, Trans. A.S.M.E., May, 1932, FSP–54–11, p. 99.

[51] From "Trends in Power-Plant Development in the United States and Europe," by C. F. Hirshfeld, Mechanical Engineering, Oct., 1929, p. 727.

TABLE LXVII

	Boiler	Superheater	Gas * Reheater	Economizer	Air Preheater
Per cent of total surface	12	6.9	17.2	17.6	46.3
Relative costs, erected	100	29.2	21.5	23.2	4.4
Relative costs per square foot, erected	100	50.7	15.0	15.8	1.1

* In this plant the steam is reheated both by the flue gases and by steam from the boiler.

of 1400 lb. per sq. in. The values given are to be considered as merely illustrative.

Fig. 776 shows the temperatures, temperature differences, and extents of surface in the different parts of a unit containing a water-cooled furnace, a boiler with tubes in two banks (water-screen and rear), interdeck superheater, economizer, and air preheater. Comparison of the drops in temperature of the gas through the different elements receiving heat only by convection gives the relations between the amounts of heat absorbed by these parts.

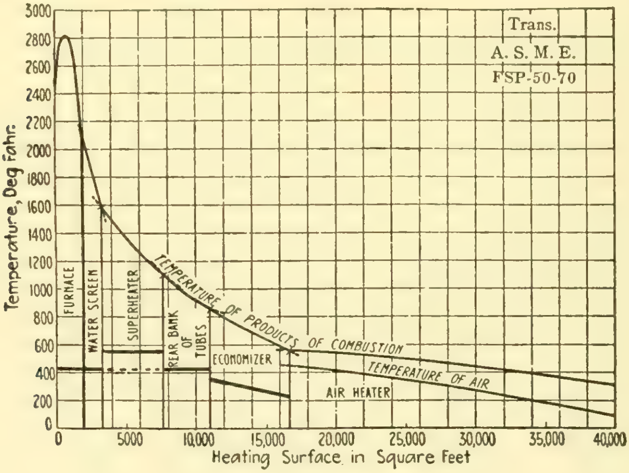

Fig. 776.

483. Boiler Materials and Construction.—(a) These factors can be considered only very briefly in this text. In general, the various parts of boilers and other pressure vessels must have adequate strength, ductility, and endurance to withstand the variable loading and the high and fluctuating temperatures to which they may be exposed under the worst conditions that can occur. The materials and construction of stationary boilers should conform to the **A.S.M.E. Boiler Construction Code,**[52] and also to any national, state, or local laws, and to any insurance and inspection regulations to which the boilers may be subject. A few of the requirements given by the A.S.M.E. Code for power boilers will be given briefly.

(b) That part of the **Code relating to materials** specifies that steel plates which are under pressure, and are also exposed to the fire or to the products of combustion, must be made of **firebox-quality,** open-hearth (O.H.) steel having a tensile strength of 55,000 to 65,000 lb. per sq. in. and a certain minimum yield point and per cent elongation. The content of carbon, manganese, phosphorus, and sulphur are assigned limits that insure ductility and avoid " hot- or cold-shortness " of the material. Steel plates for any part of the boiler that is under pressure but not exposed to fire or products of combustion may be made of **flange-quality,** open-hearth steel having the same strength as firebox steel but with less rigid limits as to chemical composition. The Code also gives specifications for materials for tubes, rivets, stays, and all other parts. Sectional headers and other pressure parts of the boiler proper, over 2 in. pipe size and subject to pressures exceeding 160 lb. per sq. in., must be of wrought steel, or cast steel made by the electric or open-hearth process. Sinuous headers may be made from round or square tubing of materials conforming to the tube specifications; malleable-iron headers may be used with pressures up to 300 lb. per sq. in.[53]

(c) The portion of the **Code relating to boiler design and construction** specifies that, when the metal in *shells* or *drums* is maintained at temperatures below 700 deg. fahr., these parts shall be designed with a factor of safety (F.S.) not smaller than 5, based on the tensile strength of the material, due allowance being made for the efficiency of riveted joints, ligaments between tube holes, and welding. However, the requirement is made that, if the metal is to be raised to temperatures in excess of 700 deg. fahr. during operation, the factor of safety must be increased with the temperature elevation so that definite working stresses shall not be exceeded at successively higher temperatures. Tabular values of the maximum allowable working stress for 50-deg. increases from 700 to 950 deg. fahr., and for plain carbon steels having various tensile strengths, are given in the Code. For example, in the case of a steel having a minimum tensile strength of 55,000 lb. per sq. in., the allowable working stress at a maximum temperature of 700 deg. fahr. is 11,000 lb. per sq. in., and the value drops to 10,000 at 750 deg. fahr., to 6750 at 850 deg. fahr., and to 4000 at 950 deg. fahr., with corresponding increases in the factor of safety.

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[52] This Code has separate sections devoted to Power Boilers, Material Specifications, Locomotive Boilers, Low-Pressure Heating Boilers, Miniature Boilers, Rules for Inspection, and Unfired Pressure Vessels. The A.S.M.E. Code has been adopted by many states and municipalities. Marine boilers for ships navigating waters under the jurisdiction of the U. S. Government must conform to the regulations of the U. S. Board of Supervising Inspectors, and also may have to meet the requirements of insurance societies such as the American Bureau of Shipping, Lloyds, British Board of Trade, and Bureau of Veritas.

[53] The 1931 Code devotes 106 pages to materials specifications, most of which are similar to those given by the American Society for Testing Materials.

When the metal of shells or drums is expected to withstand, continuously, temperatures over 700 deg. fahr., the property of the metal, known as **creep**, becomes an important factor in the design of the part. The phenomenon of creep is due to gradual changes in the structure of the steel under the influence of the continuous application of load, and manifests itself in the progressive enlargement of the part at a rate which depends directly upon the temperature elevation. For any given value of temperature, the allowable working stress must be fixed at such an amount, as noted above, that the rate of enlargement of the boiler parts will not produce undesirable results in the normal lifetime of the structure. In the case of the particular boiler parts subjected to temperatures over 700 deg. fahr., a satisfactory **creep limit** is about 1 per cent of enlargement in 10 years of operation.

In designing the pressure parts of high-pressure boilers, water walls, and superheaters, not only must the influence of creep be given very careful consideration,[54] but allowance must also be made for the stresses induced by the temperature gradients through the metal. Consideration of these matters, and of special materials suitable for use at high temperatures, is, however, beyond the province of this text.

(d) **Joints** in boiler shells or drums may be made either by riveting or welding. For riveted joints, the Code requires that the holes be drilled full size with the plates in place; or, in plates between $\frac{5}{16}$ and

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[54] Much research has been done and more is now in progress on the properties of metals at high temperatures. See "Symposium on the Effect of Temperature on the Properties of Metals," published jointly by the A.S.T.M. and the A.S.M.E. (1931); and bibliographies, reports, and other publications of these societies on the subject. See also "Creep of Steel at High Temperatures," by F. H. Norton (1929), McGraw-Hill Book Co.

$\frac{5}{8}$ in. they may be punched not to exceed $\frac{1}{4}$ in. less than full size, and in thinner plates $\frac{1}{8}$ in. less than full size, and then they are to be drilled or reamed to full size with the plates, butt straps, or heads bolted in position. In H.R.T. boilers with the shell plates exceeding $\frac{5}{8}$ in. in thickness, the circumferential lap joints exposed to the fire, or products of combustion, must have the lapping portion of each plate not over $\frac{9}{16}$ in. thick. Longitudinal joints should not be exposed to the fire or hot gases. Riveted joints are usually sealed by calking the plate edges, and the tool used for the purpose should be of such form that there is no danger of damaging the plates. In some cases circumferential joints may be sealed by fusion welding. For joints properly made by **forge welding**, the tensile strength is taken as 35,000 lb. per sq. in. When shells or drums are fabricated by **fusion welding**—i.e., the process of welding metals in the molten, or molten and vaporous, state without the application of mechanical pressure or blows—the allowed joint efficiency is 90 per cent if the welding is done and tested by the methods prescribed. After a boiler has been completed it must be subjected to a hydraulic test of $1\frac{1}{2}$ times the maximum allowable working pressure.

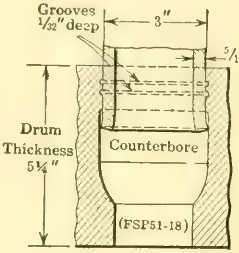

Fig. 777.—Rolled Tube Joint, 40-in. Drum, 1400 lb. per sq. in.

(e) **Boiler tubes** are sized according to their outside diameters, and their thicknesses are generally expressed in B.W.G. The Code tabulates the maximum pressures allowed for the different diameters and gages of steel and wrought iron tubes. Tube holes must be drilled full size, or they may be punched at least $\frac{1}{2}$ in. smaller in diameter than full size, and then be drilled, reamed, or finished full size with a rotating cutter. The holes may be counterbored, as in Fig. 777, when the metal is thicker than that required to get a proper tube bearing, and the sharp edges of the holes must always be taken off. The tube ends are ordinarily fastened by expanding them to a tight fit in their holes, the resulting joints having great holding power. Tube expanders used for the purpose are made up of segments or rollers surrounding slightly tapered mandrels that can be forced endways. In fire-tube boilers the tubes are firmly rolled and beaded; or rolled, beaded, and welded around the edge of the bead. In water-tube boilers the tubes must be expanded and flared not less than $\frac{1}{8}$ in. over the diameter of the tube hole; also they may be beaded or welded, and they must project between $\frac{1}{4}$ in. and $\frac{1}{2}$ in. before flaring.

(f) The A.S.M.E. Code also includes regulations covering riveted joints, dished heads, braced and stayed surfaces, tube sheets, furnace flues, finishing plate edges, riveting and calking, manholes, safety valves, water and steam gages, stop valves, blowoff valves, feed piping, fusible plugs, etc., etc. The Code should always be consulted, of course, when considering the design or construction of a stationary boiler. It is formulated with respect to usual practice but is not intended to retard the development of new designs or the use of new materials. The A.S.M.E. Boiler Code Committee will, when so requested, take under consideration new features of construction and materials. Its decisions are published in the "Interpretations of the Boiler Code," which are issued from time to time in sheet form. Incidentally, the A.S.M.E. has also prepared a separate publication, **"Suggested Rules for the Care of Power Boilers"**; and the National Electric Light Association in its Proceedings of 1925 gives suggested forms for **Instructions to Bidders** and for **Specifications for Boilers.**

484. Boiler Accessories.—(a) This term applies to (1) the **safety devices** (safety valves and fusible plugs), (2) the **indicating instruments** (such as gages for the water level, steam pressure, and draft), (3) the **control appliances** (such as regulators for the feedwater and dampers), and (4) the **servicing equipment** (such as blowoff fittings, soot blowers, and tube cleaners), which are either attached directly to the boiler or are intimately connected with the operation and maintenance of the apparatus.[55] The safety valves, water gages, steam gages, and the feedwater and blowoff fittings are always necessary equipment. The other accessories are conveniences for gaging the operation of the apparatus or for maintaining it in proper condition.

(b) **Safety valves** are of three types: (1) *dead weight*, (2) *weighted-lever*, and (3) *spring-loaded*. The A.S.M.E. Code does not permit the use of (1) or (2) because valves of this kind operate unsatisfactorily and are easily tampered with. Spring-loaded valves, with simple flat or beveled seats and discs, lift only a slight amount when functioning, and at the beginning and end of operation have long periods of simmering or leaking, which causes erosion of the seats. Commercial safety valves are of the **pop type** having specially designed valve discs and seats which cause quick opening, relatively high lift, and rapid and quick closure. The increased lifting force and more definite action are obtained (1) by extending the valve disc over a *huddling chamber* wherein the partial confinement of the escaping steam causes an additional force to act on the disc, (2) by having the escaping steam impinge on the portion of the disc in such a way as to produce an additional lifting *impulse*, (3) by having the disc deflect the jet in such a manner as to result in an upward *reaction*, and (4) by using a combination of these methods, as is done in most commercial safety valves. Fig. 778 shows one of the many possible arrangements of disc and seat making use of all three principles mentioned; and Fig. 779 is a typical arrangement of a complete safety valve.

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[55] Many of the accessories are often referred to also as *mountings*, *fittings*, or *trimmings*.

A safety valve in operation will not close until the pressure has decreased below the popping pressure by an amount, called the **blowdown**, which can be regulated by means of the adjusting ring shown in Fig. 778 (or by its equivalent in other arrangements). Raising this ring increases the pressure in the huddling chamber; this causes the valve to remain open longer, increases the range of blowdown, and decreases the effective closing force, which may prevent the valve from closing tightly. The blowdown should not be more than 4 per cent of the set pressure, and in no case less than 2 per cent. Short blowdowns mean more frequent operation and excessive wear of valve and seat. In general, the greater the blowdown, the greater is the lift and discharge capacity.

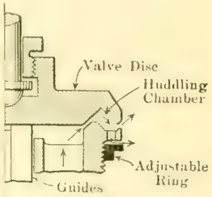

Fig. 778.—Pop Valve Disc and Seat.

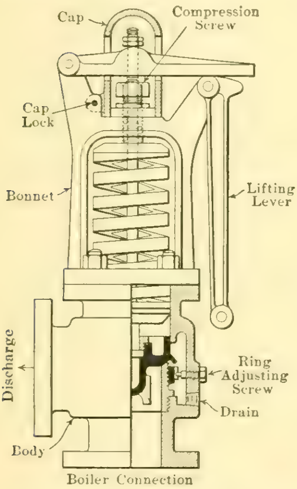

Fig. 779.—Typical Pop Safety Valve.

Two or more safety valves are required on each boiler, except very small ones, and at least one should be set to open at or below the allowed pressure. The remaining valves may be set within a range of 3 per cent above the maximum allowable pressure, and must also meet other requirements specified in the Code. Every *superheater* must have one or more safety valves of specified design, and they should be placed near the outlet.

The **maximum rated capacity** of a safety valve is the weight of steam, in pounds per hour, discharged at a pressure 3 per cent higher than that for which the valve is set to blow, and with the valve adjusted for a blowdown within the limits already given. The **total capacity** of the safety valve, or valves, on each boiler shall be sufficient to discharge all the steam that can be generated by the unit without allowing the pressure to rise more than 6 per cent above the maximum allowable pressure, or 6 per cent above the highest pressure at which the valve is set. The discharge capacity of the safety valves of an attached superheater may be included in the aggregate but not to exceed 25 per cent thereof.

The capacities of safety valves are computed according to formulas, given in the A.S.M.E. Code, based on Napier's rule, Eq. (334), with a discharge coefficient of about 96 per cent and with a valve lift corresponding to 3 per cent excess pressure. The Code permits the weight of steam generated per hour to be found by the following formula:

$$W = (C \times H \times 0.75) \div 1100 \quad . \quad . \quad . \quad . \quad (587)$$

in which

C = total weight (or volume) of fuel burned per hour at time of maximum forcing, lb. (or cu. ft.), and

H = heat of combustion of fuel, B.t.u. per lb. (or per cu. ft.).

Among **other requirements,** a safety valve should be free from shock, chattering, or any possibility of obstruction; its disc and seat should be of suitable material to resist corrosion and erosion; and it should have an adequate, open, gravity drain. The valve must be connected to the boiler as closely as possible, without the intervention of other steam connection or any valve, and must have an external lever or other device by which the valve disc may be lifted from its seat by hand. Except for high pressures, the average size of safety valve for power boilers is 4 in.

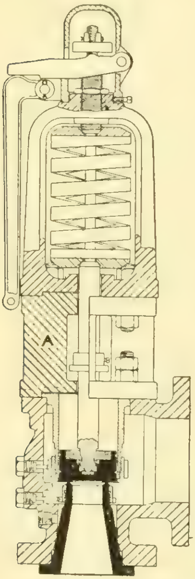

Fig. 780. — Section of Crosby Nozzle-Type Superheater Safety Valve.

For **high pressures,** the safety valves commonly have diameters of 2 in. or less. To avoid the large pressure drops resulting with the desired 4 per cent blowdown (e.g., 40 lb. when the steam pressure is 1000 lb. per sq. in.), and to reduce the frequency of operation, they may be supplemented by small warning valves or whistles, by auxiliary relief valves which may be electrically operated with less than 1 per cent blowdown, and by automatic combustion control. With **high-temperature** steam it is necessary to protect the spring from the heat of the issuing fluid, as by interposing the "cooling spool" shown at A in Fig. 780. In this design of high-temperature safety valve, the nozzle-like throat has a bushing of forged or alloyed steel which protects the valve body from the steam except when actual discharge occurs. When the valve is closed the pressure acts only on the small surface under it; but when it opens the pressure acts on the full area of the disc, thus greatly increasing the pop effect and discharge capacity of the valve.[56]

(c) **Fusible plugs,** though not mandatory, are often applied to boilers to act as signals when the water level becomes unduly low. They consist of small, hollow, bronze castings with central core of tin, as in Fig. 781, and are screwed into the boiler shell, or into a tube at the lowest permissible water level. When uncovered by the water the core melts (at from 400 to 500 deg. fahr.) and permits the escape of steam, which gives warning. The A.S.M.E. Code gives suggestions for the forms and locations of the plugs. Coatings of scale and soot affect the temperature at which the core will melt, hence the plugs should be kept clean, especially on the ends exposed to gases. These plugs are not suitable to use with working pressures above 225 lb. per sq. in. gage; and should be replaced at least once a year.

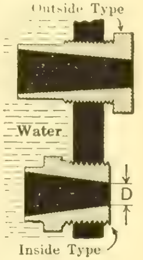

Fig. 781.—Outside and Inside Types of Fusible Plugs.

(d) The **glass water gage,** for visibly indicating the level of the water in a boiler, and the **gage cocks** or **try cocks** required by the Code may be attached either directly to the boiler or, as in Figs. 782 and 783, to a **water column** connected thereto. The lowest visible part of the gage glass should be 2 inches or more above the lowest permissible water level. The gage cocks may be of the simple hand-wheeled kind, as in Fig. 782, or of the self-closing type which are operated from the floor as in Fig. 783. The gage glass connections are fitted either with simple stop valves (Fig. 782) or with chain-operated quick closing valves (Fig. 783) for use when the gage glass breaks. The Code specifies the kinds of valves permitted and allows certain automatic shut-off valves to be used upon the water gages. Both the gage glass and the water columns must have blowoff cocks for removing sediment therefrom. Water columns frequently have floats, Fig. 783, which operate high and low water alarms, the signalling being done by steam whistles, or by electric bells, lamps, or buzzers. The floats may be used also for recording the water level on a graphical chart.

When used with **high steam pressure** the ordinary tubular gage glass lasts but one or two weeks. For such pressures it is common practice to substitute the **flat-type of water gage,** Fig. 784, consisting of two glass plates clamped flexibly into a steel housing, with mica to protect the glass from the dissolving or erosive action of the water. For pressures above 1000 lb. per sq. in. the lengths of the glass plates are limited to about 7 in. because of expansion differences, hence two or three short sections of glass are generally placed in series or at different levels to provide the necessary total length.

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[56] For further study of safety valves, see the A.S.M.E. Code; "The How and Why of Safety Valves," Power, Sept. 4, 1923, p. 357; and "Fundamentals of Safety Valve Designs," by R. J. S. Pigott, Power, April 19, 1927, p. 580.

The water level is often indicated at one or more remote points convenient for the operator by means of some form of relay or transmitting device (mechanical, hydraulic, or electric); many of these are also provided with recorders.

(**e**) The **steam pressure gages** in general use are of two kinds—**Bourdon-tube gages** and **diaphragm gages.** In the first the pressure is applied within a curved tube of oval cross-section. As the pressure increases the tube tends to straighten out and, through suitable mechanism, rotates an indicating hand or pointer which is in front of a graduated dial. In the second kind the pressure causes a corrugated disc or diaphragm to deflect and operate the indicating mechanism. Both kinds of gages are affected by heat, hence either water-entrapping pipe loops, called goose-neck siphons (see (*a*) in Fig. 715) or equivalent water-seals are placed in the pipe connecting them with the steam space in the boiler. High-pressure steam gages become inaccurate through use and should be standardized frequently.

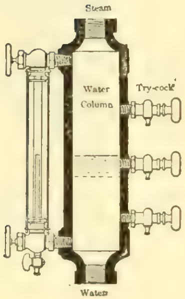

FIG. 782.—Simple Type of Water Column.

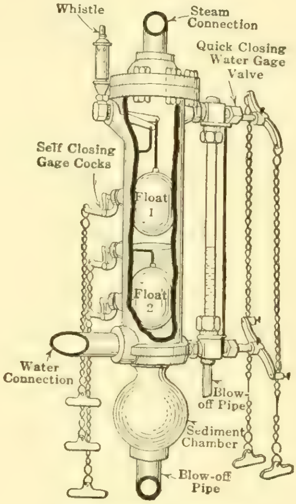

FIG. 783.—Reliance Water Column.

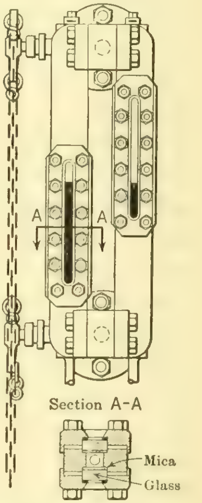

FIG. 784.—Flat-Type Water Gage.

Draft gages may be simple liquid manometers with vertical or inclined tubes, or they may be mechanical instruments which are operated by pressure-sensitive elements such as weighted diaphragms, bell floats, or sylphon bellows. The latter have thin deeply corrugated cylindrical shells which are accordion-like in action.

Other instruments of value in gaging the operations of boiler units include those for measuring gas composition, feedwater pressure and temperature, steam flow, water flow, fuel feed, and gas flow and temperatures. The simple indicating instruments may be supplemented by, or replaced by, *recording instruments* or meters which give instantaneous, continuous, and permanent records of the quantities measured. These instruments may be located near the boilers or in more remote places, such as in a central control room or in the superintendent's office.

(f) The **bottom blowoff valves and connections** are subjected to very severe service. The blowoff lines must be of extra heavy wrought iron or steel pipe, or thicker pipe, of size from 1 in. to $2\frac{1}{2}$ in., must be protected from furnace heat (e.g., Fig. 715), and must be free to expand and contract. A large water-tube boiler will have several blowoff connections. Each blowoff pipe must have either two slow-opening valves, or one slow-opening valve and cock, all of extra heavy construction; or a master valve may be placed on a common blowoff pipe and only one valve be used in each individual blowoff connected thereto.

The outer or *blowing valve* controls the discharge, and it should be of the slow-acting type so as to avoid the possibility of producing water hammer when opening or closing it; also it should furnish a free passage for the scale and mud, have no pockets in which the sediment can collect, operate with the minimum of erosion, open easily, and be free from seat cutting. The intervening valve or cock is a *sealing valve*, which is depended on to keep the line tight and to permit repairing or replacing the blowoff valve without shutting down the unit. In order to maintain its tightness this valve or cock should be opened first and closed last, i.e., it should be manipulated only when no flow is occurring. When circumstances permit, the blowoff conditions can be made less severe by reducing the boiler pressure before blowing.

Two of the many different designs of blowoff valves are shown in Figs. 785 and 786. In the first the disc is reversible, has two seating surfaces that can be renewed, and fits the ring above the seat in such a manner as to prevent the passage of large particles of scale during the wire drawing which occurs when the valve is just being seated or unseated. Fig. 786(*a*) shows a seatless blowoff valve in which the packing around the plunger is compressed when the shoulder S comes in contact with the end F of the gland at the end of closure. At (*b*) and (*c*) of the same figure are shown tandem arrangements of blowoff valves, the former having a sealing valve of the quick-opening swing-gate type, and the latter having a conical cock.

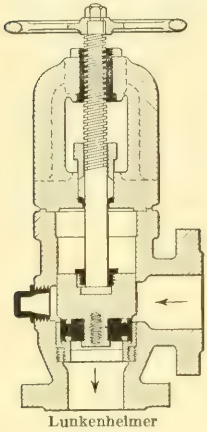

FIG. 785.

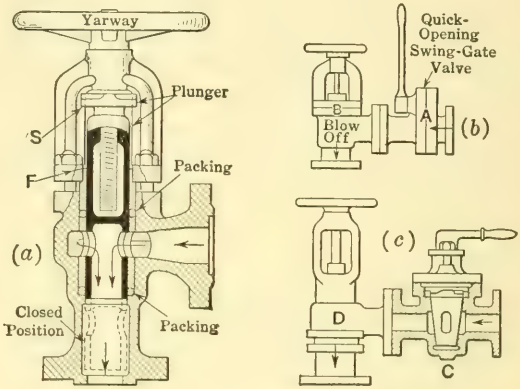

FIG. 786.

A Triplex blowdown combination for high pressures (e.g., 1400 lb. per sq. in.) is shown in Fig. 787. It consists of two main seatless valves in series, and a 1-in. seatless valve connected between their annular spaces. The actual *blowing down* is done by means of the easily operated and less troublesome 1-in. valve, after first opening the lower large valve. The upper valve need be opened only for *draining* the boiler, which is done at a low pressure.

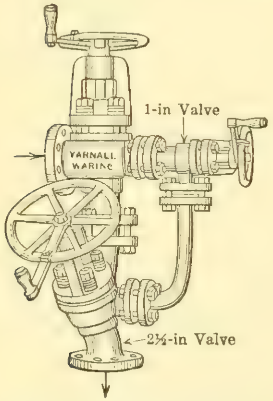

FIG. 787.—Triplex Blowdown Combination (Yarway).

In addition to the bottom blowoff, boilers using poor water are sometimes equipped with **surface blowoff devices** for skimming off the upper layer of water in the apparatus so as to remove scum, grease, and floating or suspended particles of foreign material. City ordinances require all blowoffs to be discharged into tanks to allow cooling before delivery to sewers.[57]

(g) **Automatic feedwater regulators** are used in practically all modern central stations and in many industrial plants. Their purpose is to supply the boilers with water at the same rate as it is being converted into steam, and thus to maintain the water levels substantially constant, without the close attention that is necessary with hand regulation. The most common types are operated either by floats or by the thermostatic action of metal or of confined fluid, and they depend for their primary control on variations in water level.

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[57] Continuous blowdown is discussed in Chapter XLV.

In the **float type** of regulator, the variations in the water level cause the float to operate a balanced feed valve, through suitable intervening linkage. Many different designs of this type of regulator are on the market.

The **metallic thermostatic type** of regulator is exemplified by the Copes feedwater regulator, a simple arrangement of which is shown in Fig. 788. Changes in water level alter the proportion of the inclined tube 2–3 exposed to steam, the water in the rest of the tube being at a constant low temperature only slightly above that of the room. The upper end of the tube is free to move and is connected to a bell crank which, through suitable linkage, controls the position of the feed valve so that its opening is substantially proportional to the expansion of the tube.

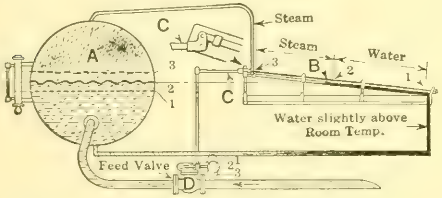

FIG. 788.—Elements of Copes Feedwater Regulator.

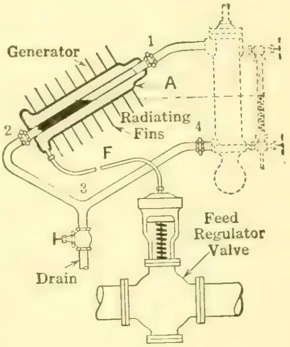

FIG. 789. — Fluid Thermostatic Feedwater Regulator.

A **fluid thermostatic type** of regulator is illustrated in Fig. 789. The tube 1–2–3–4 is connected to the steam and water spaces in the boiler or water column in the manner shown, and the water in it is at a temperature much below that of the steam. The annular space *A* around the upper part of this tube constitutes a generator which is connected by tube *F* to a spring-balanced diaphragm, sylphon, or equivalent device, which actuates the feed valve. The generator and its connections to the valve constitute a closed system containing the thermostatic fluid, usually water. Variations of water level in tube 1–2 alter the relation between the rate at which the fluid in the generator receives heat from the steam in that tube and loses it through the external fins. The thermostatic changes in the fluid regulate the amount of opening in the feed valve.

Most feedwater regulators that are actuated solely by changes of water level in the boiler continue for a time to supply water after a sudden increase in pressure or drop in load causes the steam-charged mass of water to contract, and they generally cease feeding when there is a sudden demand which makes the mass swell because of more rapid ebullition. Such regulation is unsuitable when there are widely fluctuating loads and the boiler is operated at a high rate. (Usually all the water can be evaporated out of high intensity boilers in less than 2 minutes if the feed is shut off while carrying maximum load.) For these conditions the regulators are often supplemented by sensitive feed controls which are operated generally by variations in the steam flow. Many of these auxiliary devices regulate the drop in water pressure through so-called excess pressure valves which are placed ahead of the usual feedwater regulator valve.[58] In one regulator (the Bailey) the primary control depends upon the differential between the rates of flow of the steam and water, the water level element being used merely to hold the level within desired limits.

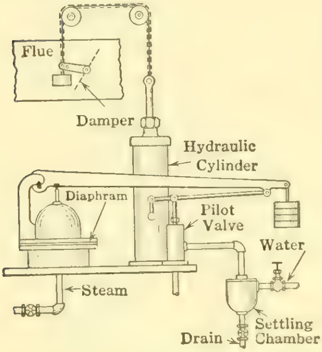

Fig. 790.—Damper Regulator.

(h) A **damper regulator** is a device for automatically controlling the boiler- or stack-damper. Most of those for hand-fired, natural-draft boilers are actuated by variations in the boiler pressure. In this type of regulator the steam pressure may act on a spring-balanced piston which is connected directly to the damper through suitable mechanism, or it may act on a weighted diaphragm, as in Fig. 790, Bourdon tube, mercury column, or similar device, which indirectly operates (through a relay system) a hydraulic or compressed-air cylinder or an electric motor for moving the damper. A drop in steam pressure causes the damper to open wider, thus accelerating combustion, and a pressure-increase reduces the damper opening. Under sudden fluctuations in steam demand a continued hunting between the variations in steam pressure and the damper movement may occur unless the design of the regulator has special features to avoid such action. Damper regulators may also be actuated by variations in the draft pressure in the furnace or by fluctuations in the rate of flow of steam in the delivery pipe. When boiler units are fired by stoker, pulverized coal, oil, or gas, and mechanical draft is employed, the regulation of the dampers is usually coordinated with the operation of an automatic combustion control system.

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[58] Power Plant Engineering, Jan. 15, 1928, p. 117. Proc. N.E.L.A., 1927, pp. 946 and 970; 1929, p. 1315.

(i) **Scale and soot removal** from the heating surfaces of steam generating units is necessary at suitable intervals of time, the economic length of which depends on (1) the character of the deposit, (2) the rapidity of its formation, (3) the cost of removal, and (4) the value of the improvement in efficiency and capacity resulting therefrom, the improvement being indicated by the reduction in the temperature of the exit gases.

Soot found in such apparatus may have little resemblance to lampblack and often is not even black. Coatings of such material can be removed by means of brushes, scrapers, steam lances, and similar appliances, and by washing. Light, flocculent deposits are commonly removed by using blasts of steam from **hand-operated lances,** or from permanently installed **mechanical soot blowers.** The latter, see Fig. 791, consist primarily of tubular elements having a multiplicity of steam nozzles, the necessary piping and valves, and means for rotating the elements—usually by chains manipulated from the floor, but sometimes electrically, the operation being controlled either at the boiler or from a central point. The issuing steam jets should not impinge on the tubes in such a manner as to cause erosion, and the elements in the hotter positions must be protected by barriers or sheathing, or be made of special non-warping alloys which withstand the heat and the erosive and corrosive action of the ashes and cinders. At points where the heat is extremely intense it is necessary to dispense with soot blowers altogether to avoid excessive replacement expense. Steam blasts are not always suitable for removing tar, gum, slag, and other adhesive deposits, which generally must be dislodged by other methods.

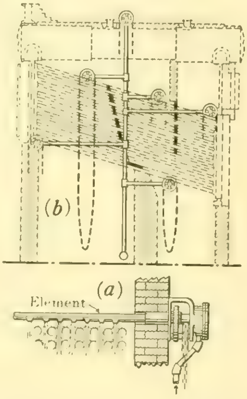

Fig. 791.—One Arrangement of Soot Blowers.

The steam consumption of soot blowers is from 0.2 to 0.6 per cent of the total weight of steam generated by the unit, being greater in old equipment and with units fired by pulverized coal.[59] The steam used by soot blowers may be minimized by properly reducing (1) the steam pressure used, as by placing an orifice in the steam supply pipe, (2) the frequency of operation, (3) the number of blows at each cleaning, and (4) the time between the opening and closing of the supply valve and the actual use of the blower. Some designs have mechanisms in the heads

[59] For data on consumption, see Proc. N.E.L.A., 1927, p. 956.

of the elements for automatically controlling the steam blast so as to eliminate waste.

(j) **Internal coatings,** deposited by the water on heating surfaces of steam generators, may be removed by washing, when soft and mud-like; but when hard scale is formed the cleaning is accomplished by scraping, or in the case of tubes, by the use of **mechanical tube cleaners,** of which there are two general types: (1) the **hammer type** and (2) the **revolving cutter type.** The cleaner heads of such appliances are usually attached to flexible pipes, through which is fed the fluid which operates them and by means of which they are pushed through the tubes being cleaned. The *hammer cleaner,* one form of which is shown in Fig. 792, may be operated by steam or compressed air. It has a vibrating head which subjects the tubes to several hundred shocks or raps per minute, by which brittle scale is cracked and jarred loose from either the internal surfaces of water tubes or the external surfaces of fire tubes. This type of cleaner should be operated in such manner as to avoid swaging the tubes to larger diameter or loosening the expanded joints. The *revolving cutter* type, exemplified in Fig. 793, is applicable only to water tubes and is furnished with various types of cutters to suit the different kinds of scale encountered. The cutter head contains small paddles or a turbine which is driven by steam, compressed air, or water under pressure.

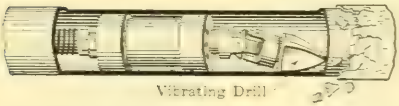

FIG. 792.—One Form of Hammer Type of Tube Cleaner.

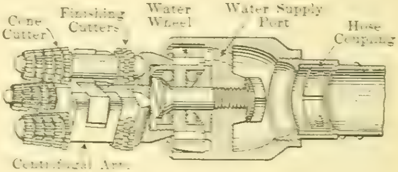

FIG. 793.—Typical Revolving Cutter Type of Tube Cleaner.

ADDITIONAL REFERENCES

"The Benson Boiler," by F. Ohlmuller, Combustion, Jan. 1933, p. 11.
"Schmidt High Pressure Boilers," by J. Cunningham, Combustion, Feb. 1933, p. 21.
Special Boiler Number of Power Plant Engineering, Jan. 1934.
"Some Experiences with the Benson Boiler," Combustion, March, 1934, p. 25.
"The Velox Steam Generator," by M. G. S. Swallow, Engineering, Apr. 20, 1934, pp. 469, 526 (431).
"The Velox Steam Generator," by A. Meyer, Combustion, May, 1934, p. 28.
"Load Regulation by Pressure with the Benson Boiler," Combustion, June, 1934, p. 25.
"Loeffler Boiler Tests," by E. Josse, Engineering, June 29, 1934, p. 722.
"Loeffler System of Steam Production," by S. McEwen, Inst. Mech. Eng. Nov. 9, 1934; Engineering, Nov. 16, 1934, pp: 535, 546.
"The Trebovoice Power Station" (Loeffler Boiler), Engineering, Nov. 2, 1934, p. 457.
"The Benson Boiler and Its Application," by F. Hodgkinson, Combustion, March 1935, p. 21.

CHAPTER XL

SUPERHEATERS, REHEATERS, AND DESUPERHEATERS

485. Superheater Types, Arrangements, and Characteristics.—(**a**) Superheaters are classified broadly either as (1) *detached superheaters* or (2) *integral superheaters*; detached ones are usually *separately fired,* but sometimes utilize waste heat from industrial furnaces.

Separately fired superheaters, although requiring special settings, additional space, and frequent attention, may be used to advantage (1) for increasing the capacity or efficiency of an old plant, (2) for superheating the steam exhausted from steam hammers or reciprocating engines before it is used in a low-pressure turbine, (3) for superheating a portion of the steam that is required at a higher temperature than the rest, or (4) for giving flexibility of temperature control.

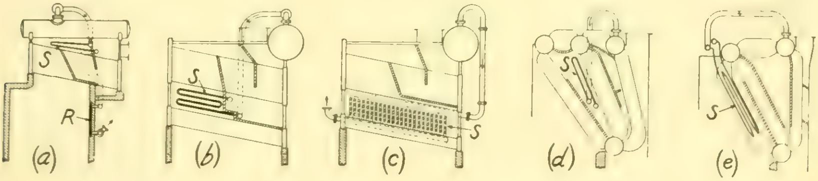

Fig. 794.—Typical Locations of Integral Superheaters.

The **integral type of superheater,** being installed within the setting of the steam generator, requires no special attention, and is the kind commonly used in power plants; the discussion which follows will be confined to this type.

(**b**) The **location** of an integral superheater within the steam generator determines to a large extent its further classification, the amount of surface required, and the materials and construction to be used in its structure. The superheaters shown at S in (*a*) and (*d*) in Fig. 794 are of the **interpass** convection type; such a superheater is located between the first and second gas passages of the boiler and is therefore applicable only when moderate steam temperatures are required. The superheaters in (*b*) and (*c*) are of the **interdeck type**; they are located between upper and lower banks of boiler tubes, and though *primarily of the convection type,* also receive a portion of heat by *radiation* if but a few boiler tubes are interposed between them and the furnace. The tubes in (*b*) are arranged longitudinally, and in (*c*) transversely, with respect to the boiler tubes. A **radiant superheater** is shown at *R* in (*a*), installed in the furnace wall and connected in series with the convection superheater *S*. The superheater shown in (*e*) is of the **inter-tube type**; the loops are placed between the boiler tubes in the first pass of the gases and in such position that they receive heat both by convection and by radiation.

In locomotives the superheater tubes are commonly placed in the boiler tubes or flues, as shown in Fig. 720 (p. 675); they may be similarly placed in Scotch marine and other large-tube boilers; or they may be in the combustion space.

In general, the nearer an interdeck or inter-tube superheater is to the furnace, the smaller need be the surface provided, and the higher can be the steam temperature produced, but the better must be its materials, construction, and condition. The amount of boiler-heating surface between the interdeck superheater and the furnace should be such as to give a suitable temperature difference (e.g., 300 deg. fahr.) between the gas and steam. Temperature gradient curves such as those in Figs. 771 and 776 may be used in locating the position of the superheater with respect to the other heat-absorbing surfaces of the unit.

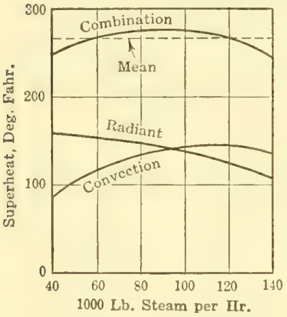

FIG. 795.—Characteristic Curves of Superheaters.

(**c**) The **performance characteristics** of convection and radiant superheaters are shown by the curves in Fig. 795. The flame temperature to which a *radiant superheater* is exposed generally increases with additional load less rapidly than does the steam output; hence the characteristic curve for this type of superheater shows the degree of superheat as falling with increased output. On the other hand, with greater forcing of the unit, a *convection superheater* is swept by gases in proportionally greater amount and of higher temperature, and the rate of heat transmission is also increased because of the higher gas and steam velocities; hence, for this kind of superheater, at first the superheat curve rises rapidly with increase in output as shown by the curve. However, when very high flow rates are approached, the rate of increase in superheat will become less, and at extremely high loads the steam temperature will fall; this is because (1) the overall rate of heat transfer from the gas to the steam does not increase as rapidly as the rate of steam flow,[1] and (2) increased moisture is carried by the saturated steam.

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[1] See Sect. **488**(**b**), p. 752.

The moisture entrainment can be avoided by placing suitable steam separators or dryers in the steam drums of the boiler.

(d) Fairly **uniform temperature** of steam can be obtained by placing *radiant and convection superheaters in series*, the flatness of the resulting characteristic curve being dependent mainly on the proportions and locations of the respective superheaters with respect to the other parts of the unit. Interdeck and inter-tube superheaters give flatter curves than do interpass superheaters, the flatness being dependent on the ratio of the heat received by radiation from the furnace to that transmitted from high-temperature gases. Uniform temperature at the superheater outlet, however, is not always desirable; when the steam is used in a high-pressure turbine which is regulated by throttling and has small blade clearances, the temperature at the turbine, rather than that at the superheater, should be maintained constant. Thus, to offset the decrease in temperature due to throttling, the superheater should give higher temperatures to the steam under light loads than under heavy ones, a requirement that is met by the radiant type. With low boiler pressure, the decrease in temperature with throttling is less pronounced and the characteristic curve should therefore be flatter than with high pressure.

(e) **Temperature regulation** of superheated steam can be effected (1) by using a desuperheater in combination with a superheater, which must then be over-size; (2) by using a by-pass damper to change the amount of furnace gas passing over a convection superheater; (3) by varying the CO_2 percentage so as to change the temperature of the gas; (4) by recirculating through a convection superheater gases withdrawn from some other part of the unit; and (5) in W-Stirling or double-set units, by having different amounts and kinds of superheater surface on the two sides of the unit and varying the apportionment of furnace gas to them through damper adjustments on the two boiler outlets.[2]

483. Convection Superheaters.—(a) Superheaters of this type are of tubular construction, usually with either **hairpin loops** or **zigzag multiple loops** extending between steel headers, which are either forged to shape or made of seamless tubing; and the tubes may be either bare or covered. By using fusion welding, the tubes of superheaters, reheaters, and similar apparatus can be made of any desired continuous length. Typical applications of convection superheaters are shown in Fig. 794, and details are indicated in Figs. 796 to 800 inclusive.

(b) A **hairpin-loop interpass superheater** for obtaining moderate steam temperatures is shown in Figs. 725 and 796, and consists of flexibly supported U-bends of 2-in. seamless cold-drawn tubes that function in parallel; these tubes are expanded into wrought-steel rectangular headers, which contain covered handhole openings opposite each group of four tube ends. Steam from the superheater is delivered to the center fitting *A* to which is attached the superheater safety valve and the delivery pipe or valve.

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[2] "Compensating Superheaters," by P. W. Thompson, Power, April 26, 1932, p. 620.

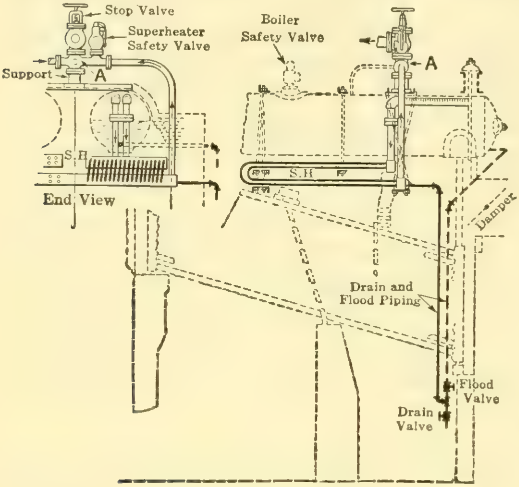

FIG. 796.—Hairpin-Loop Superheater (B. & W.).

(c) In the **multiple-loop superheater,** shown in Fig. 797, the tubes are expanded into circular headers, and a Y-cleanout fitting is provided, although not always necessary. The bends are formed in such manner that neither the passage area nor the wall thicknesses are decreased. Applications of similar superheaters are illustrated in Figs. 733, 734, and 735.

In the multiple-loop superheater shown in Fig. 798, the tubes constitute detachable units which are clamped to the headers with spherical or ball joints. Each of the return bends is formed by forging together by a special process the ends of two parallel tubes. The form of the bend is shown in detail at (*a*) in the figure, and is such as to provide

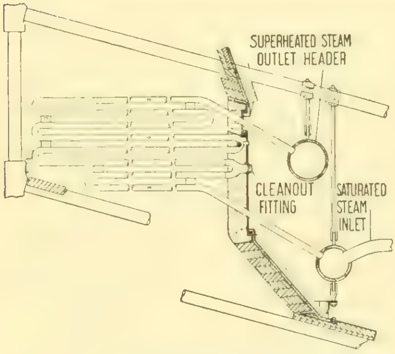

FIG. 797.—Multiple-Loop Superheater (B. & W. Interdeck).

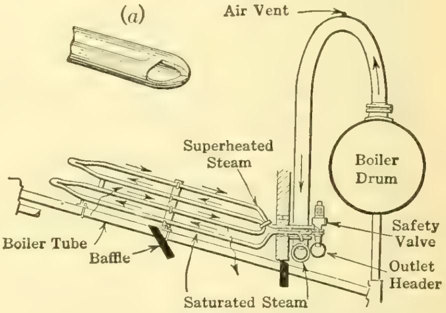

FIG. 798.—"Elesco" Multiple-Loop Superheater (The Superheater Co.)

increased wall thickness and passage area at the turn. Other applications of such superheaters are shown in Figs. 746 and 748.

(d) A **sleeve-protected superheater tube** with extended surface is shown in Fig. 799. The tubes, usually 2 or 4 in. in diameter, are covered with cast-iron sleeves which are reamed to gage and shrunk on. The extended surfaces of these sleeves are varied in contour and extent to suit the different conditions encountered. The sleeves add somewhat to the cost of the superheater; but they control the ratio between the areas of the external and internal surfaces; they protect the steel tubing from pitting, burning, or rusting; they also have a steadying effect on the steam temperature, and prevent damage from overheating during the process of steam raising. Pipe cores, or distributing plugs, can be added to apportion the steam equally among the various elements, and to cause the steam to flow at high velocity in a thin film in intimate contact with the heating surface.

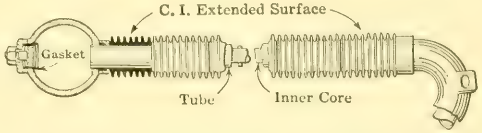

Fig. 799.—Sleeve-Protected Extended-Surface Superheater Tube (Foster Wheeler Corp.)

(e) The **design of the convection superheater** should be such that the **velocity of steam flow** through the tubes will be sufficient to prevent them from becoming overheated; for any given pressure the higher the temperatures of the steam and gases, the greater must this velocity be; and, according to Orrok,[3] this velocity should not fall below 40 to 50 ft. per sec. Usually many tubes in parallel are required; and with such construction, increased steam velocity with a given extent of surface can be obtained by using (1) fewer and longer tubes in parallel, (2) smaller tubes with proportionally greater length, and (3) cores in tubes. High velocities with the hairpin construction can be obtained by partitioning the headers, as in Fig. 800, in such manner as to cause groups of the loops to function in series.

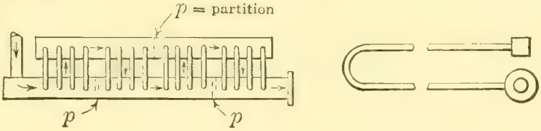

Fig. 800.

The **drop in pressure** through the tubes increases with the velocity of flow and necessitates raising the boiler pressure a like amount above the superheater delivery pressure. The drop in pressure through the superheater ranges from 1 to 15 per cent of the boiler pressure, depending upon the rate of operation and the design of superheater; the usual drop is from 2 to 5 per cent.[4]

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[3] "High Pressure Steam Boilers," Trans. A.S.M.E., FSP-50-32, 1928, p. 52.

Proper steam distribution among tubes operating in parallel is also of great importance and is obtained by (1) using headers of larger cross-section compared to the passage area through the tubes, (2) having the steam enter the first header and leave the last one at opposite ends, (3) supplying the saturated steam to the first header at numerous places equally spaced along its length, and (4) using equalizing ferrules, baffles, or cores in the passages.

The **stress in the tube wall** of a superheater is due not only to the steam pressure in the tube, but also to the difference in temperature between the outer and inner layers of wall metal; and for very high temperatures of the steam special and more costly material for the tubes is necessary. Unless very high temperatures are required, the use of special materials may be obviated by (1) having the saturated steam enter the tubes located in the hottest zone, (2) increasing the steam velocity, (3) using thinner, hence smaller, tubes, (4) preventing the "laning" of the furnace gases, and (5) avoiding secondary combustion adjacent to the tubes. The importance of reason (1) given above is usually sufficient to justify the use of *parallel-flow* superheaters instead of *counter-flow* ones even though the latter would require less surface and would give more nearly uniform temperature under varying loads.

The **material** forming the walls of superheater tubes must be kept within a safe creep limit and must be able to resist oxidation. Suggested limits of the more normal **metal temperatures** that might be used with different materials are as follows:[5]

900 deg. fahr. for 0.15 per cent carbon steel, or 1000 deg. fahr. when calorized;
925 deg. fahr. for 0.40 per cent carbon steel, or 1050 deg. fahr. when calorized;
and 1300 deg. fahr. for alloy steel having 18 per cent chromium and 8 per cent nickel if the fuel used contains 1.5 per cent sulphur or less, but only 1100 deg. fahr if the sulphur is as high as 2.5 per cent.

The **supports** for superheater tubes should be durable and sufficiently flexible to provide for changes in form of the superheater with temperature variations; the supporting members should either be protected from exposure to high-temperature gases or should be of material that will withstand such exposure.

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[4] See Proc. N.E.L.A., 1931, p. 756.

[5] Proc. N.E.L.A., 1931, p. 787.

The **headers** of superheaters should be located outside of the setting, or in the cooler regions therein, or else they should be protected by baffles or screens.

(f) The **operation of convection superheaters** with safety requires that the tubes be not subject to overheating during the periods of (1) starting, (2) normal operation, and (3) emergency. Formerly it was thought necessary to flood the superheater (as by the means shown in Fig. 796) when starting or when the demand suddenly ceases; but now this is ordinarily not necessary. When firing up a unit, sufficient protection is afforded the tubes by opening the drain valve on the superheater outlet so that a small quantity of steam will pass through them. To protect the superheater from damage when the demand for steam suddenly ceases, a **safety valve** is placed on the superheater outlet (see Fig. 796); the capacity of this valve should be sufficient to assure an adequate flow of steam through the elements even though there is delay in reducing the rate of firing. External deposits may reduce the steam temperature 75 deg. fahr. or more, hence the necessity for frequent soot blowings.[6] Internal scale also decreases the steam temperature and increases the temperature of the tube wall. At all times the tubes should be kept free from scale and soot. Interior surfaces will remain clean if the steam flows through the tubes with sufficiently high velocity and has less than 1 per cent moisture upon entering them.

487. Radiant Superheaters.—(a) Superheaters of this type, being installed in the furnace wall and exposed directly to the source of heat, (1) are relatively small; (2) do not interfere with the arrangement of the boiler tubes; (3) do not affect the draft; (4) are easily placed in old boiler settings, which may not have space for adding convection superheaters; and (5) may be coupled in series with other superheaters which have insufficient heating capacity or unsatisfactory temperature-load characteristics. Radiant superheaters should be credited with the cost of the furnace wall replaced by them and with the wall maintenance obviated. When installed in a refractory wall, the forcing capacity of the furnace is increased.

Radiant superheaters must be designed, constructed, installed, and operated with care because of the very severe conditions to which they are subjected; especial attention should be given to (1) the distribution and flow of steam through the elements, (2) the internal stresses resulting from heating the parts mostly on the side exposed to the furnace, (3) the maintenance of scale-free surfaces, and (4) the care exercised in putting a unit on, or taking it off the line. If, when used in combination with convection superheaters, the saturated steam is brought to the radiant elements instead of to the convection ones, the former are less liable to become overheated, and have greater rates of heat transmission than would otherwise be the case.

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[6] Some stations have soot blowings only once a week, others once a day, and others from 1 to 6 times per day. See Proc. N.E.L.A., 1931, p. 762.

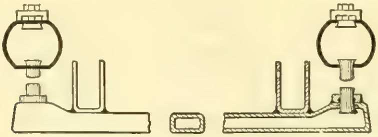

Fig. 801.—Forged-Steel Radiant Superheater Element. (Foster Wheeler.)

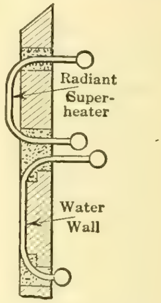

Fig. 802.—Tubular Radiant Superheater.

(b) In **early designs** of radiant superheaters the elements consisted of cast-iron blocks shrunk onto steel tubes. Later, cast-iron elements were substituted, and still later flat forged-steel ones were introduced. An element of this last type, shown in Fig. 801, has a rectangular cross-section with walls about $\frac{3}{4}$ in. thick. The elements can be arranged side by side, either horizontally or vertically, so as to form a continuous flat wall; they are connected to the headers by nipples, opposite which are handholes; their method of support permits them to expand and contract independently from the boiler; and they are backed with insulating material which in turn is enclosed in an air-tight housing, normally formed of sheet-steel panels. Superheaters of this type are still used.

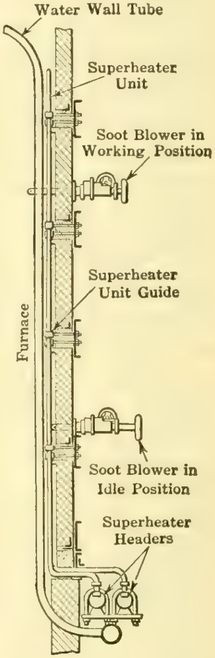

Fig. 803.—Screened Radiant Superheater. (The Superheater Co.)

More **recent designs** of radiant superheaters have used simple round tubular elements, usually 2 in. in diameter, the tubes being either exposed directly to the furnace or being placed behind water-wall tubes which act as partial screens; Fig. 802 shows an installation of the first type. An example of the screened arrangement is given in Fig. 803, the superheater elements or units being of the Elesco construction, the loops being inverted; the units are held in position in such manner that they are free to expand and contract, and they are cleaned by means of soot blowers which can be withdrawn from the furnace when not in use. Adjustment of superheating capacity can be readily effected by placing screened tile between the water wall and the superheater units.

488. Heat Transmission in Superheaters.—(a) Heat is transmitted to the tubes of a superheater of any type from the hot solid particles, gases, and surfaces by *convection* and *radiation*; this energy is transferred through the metal tube by *conduction* and thence to the steam by *convection*.

(b) For **convection types of superheaters** the resistances to the flow of heat from the gas to the steam are due to the *gas film*, the metal *tube wall* with its *deposits*, and the *steam film*, as shown by Fig. 492 (p. 282) for a similar case except that the heat is absorbed by water instead of steam. The *gas-film resistance* to heat transfer is much greater than the sum of the resistances of the metal tube and the steam film when a convection superheater has any appreciable flow of steam through it; and at high rates of operation the steam-film resistance becomes almost negligible in comparison with that of the gas film; hence the gas-film resistance is the chief factor in controlling the overall coefficient[7] of heat transfer in convection superheaters and gas reheaters.

The gas-film coefficient of heat transfer for such equipment as convection superheaters, air heaters, and economizers depends upon a large number of variables; and probably one of the best attempts to give an equation for this coefficient is that of Chappell and McAdams, as expressed by Eq. (420) (pp. 271 and 272). A careful analysis of the application of this equation shows that for any given arrangement, spacing, and size of tubes, the *mass velocity* (w/A) of the gas is the controlling term for various rates of operation.

An example will assist in showing the importance of the mass velocity of the gas. Suppose that the range of operation of a given superheater involves a threefold increase in the mass velocity, w/A, with a corresponding increase in the *mean absolute temperature*, T_f, of the gas and the tube surface from 1200 to 1400. Then if the exponent for w/A in Eq. (420) be taken as 0.7 the threefold increase in w/A will cause $(w/A)^{0.7}$ to become 2.16 times its former value, while $T_f^{1/3}$ becomes only 1.053 times its former value.

This example is a typical one and the values chosen are reasonable; it shows clearly that the 5 per cent increase in the film coefficient due to the greater value of T_f is insignificant compared with the corresponding 116 per cent increase in this coefficient due to the greater value of w/A.

Some *actual temperatures* pertaining to a convection type of superheater[8] are shown by the curves in Fig. 804; and the same values are also given in Table XLVII (p. 454). These test results show that the temperature of the steam leaving the superheater begins to drop in a typical manner after the flow rate becomes very high. The reasons for this drop in temperature have already been briefly stated in Sect. **485**(**c**); but now the reason why the overall rate of heat transfer does not increase as fast as the steam flow needs to be considered.

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[7] The equations for this coefficient, in general, are given on p. 283.

[8] In Unit No. 22, Calumet Station, see Fig. 545 (p. 453).

The increase in the temperature of steam (Δt_s) in passing through a convection superheater may be expressed by the following equation:

$$\Delta t_s = \frac{q}{w_s c_p} = \frac{UA\theta_m}{w_s c_p} \quad . \; . \; . \; . \; . \; . \; . \quad (588)$$

in which A = area of superheater surface exposed to the hot gases, in sq. ft.;
c_p = the mean specific heat of the steam;
q = overall rate of heat transfer, in B.t.u./hr.;
U = overall coefficient of heat transfer, in B.t.u./hr./deg.fahr./sq. ft.;
w_s = steam-flow rate, in lb. per hr.;
θ_m = mean temperature difference between the gas and steam.

For any given superheater, the value of A is fixed, and c_p is also nearly constant because the pressure and temperature of the steam do not vary over a wide range. Consequently the rise in the steam temperature will depend on $U\theta_m/w_s$. When the steam flow becomes larger the rate of firing must be increased, with the result that the temperature of the gas passing over the superheater surface increases, but in smaller proportion than the steam flow, as shown in Fig. 804; hence θ_m will increase somewhat because the steam temperature rises only a small amount. Furthermore, when the steam flow increases, the flow of gases also increases nearly in direct proportion; but the increase in the gas flow will cause the overall coefficient, U, to increase only as some power of the mass velocity of the gas; *this power is appreciably less than unity* and lies between 0.6 and 0.8, as indicated by Eq. (420) (p. 271). Consequently, when the steam-flow rate through the superheater is increased, the product of U and θ_m, in Eq. (588), will at first increase faster than w_s thus causing Δt_s to become greater; but at higher flow rates

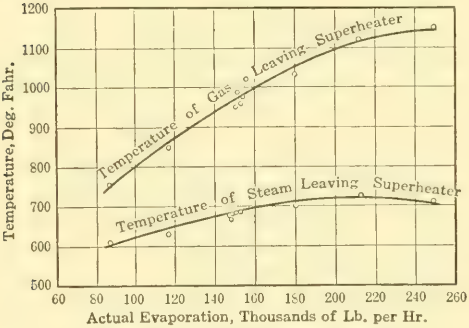

FIG. 804.—Convection-Superheater Temperatures.

the value of $U\theta_m$ will not increase as rapidly as w_s, and thus Δt_s will eventually begin to drop. This drop is further intensified by the greater tendency for the boiler to deliver to the superheater steam containing more moisture as the rate of operation becomes higher.

A *rational determination* of the rate of heat transmission from the gas to the steam in convection superheaters cannot be made without complete information concerning the mass velocities of the gas and steam, the temperatures of the gas and steam entering and leaving the superheater surface, the material and size of the tube, and the nature of any deposits. With so many factors involved, one naturally expects to find a wide variation in the rate of heat transfer in the various convection superheaters in use; some operating results [9] for units installed in 1928 and 1929 show values of this rate, in B.t.u. per hour per square foot, ranging from 4000 to 14,000 for average operating conditions, and from 7000 to 20,000 as maximum values.

(c) With **radiant-type superheaters** heat is transmitted to the tubes by radiation from the hot furnace walls, from the carbon dioxide, water vapor, and solid particles in the products of combustion, and from the fuel bed in hand- or stoker-fired furnaces. The rate of heat absorption at the superheater surface depends upon the temperature and emissivity of the sources of radiant energy, the solid angle throughout which the superheater surface " sees " these sources, and the temperature and emissivity of the superheater surface. With complete information concerning these variables, a rational determination of the rate of heat absorption may be made by the methods of Sect. **337**.

The rate of heat absorption in radiant superheaters is much greater than that in convection types; but data concerning the limiting values of this rate for radiant superheaters are meager. However, many forged-steel superheaters with walls $\frac{3}{4}$ in. thick have functioned satisfactorily at a rate of heat absorption of 45,000 B.t.u./hr./sq. ft. with a final steam temperature of 700 to 750 deg. fahr. Heat transfer rates with thin tubes seldom exceed 60,000 B.t.u./hr./sq. ft.; and, according to Drewry,[10] the practical limit is about 75,000.

The temperature characteristic of a radiant superheater depends chiefly upon its *position* relative to other heat-absorbing surfaces, and also relative to the flames in the furnace. The flame position can be easily changed to a considerable extent in a gas or powdered fuel installation and thus the steam temperature from a radiant superheater in such a furnace can be adjusted somewhat. Usually, however, the temperature of the steam from a radiant superheater decreases at high rates of operation as indicated by the curve in Fig. 795. This is chiefly because the fourth power of the absolute flame temperature is the main factor affecting the rate of heat absorption per unit weight of steam flowing through the superheater; and this factor does not generally increase as fast as the steam output. Thus, for example, by using the curves in Fig. 708 and assuming a furnace with a fraction cold of 40 per cent, powdered coal, and 20 per cent excess air at 70 deg. fahr., the mean temperature of the gases in the furnace (t_U) is seen to be 2200 deg. fahr. for an energy release rate of 15,000 B.t.u./hr.; and for twice this rate t_U is 2500 deg. fahr. Then by reducing to absolute temperatures, $t_{U_1} = 2660$ and $T_{U_2} = 2960$; hence $(T_{U_2}/T_{U_1})^4 = (1.112)^4 = 1.53$, which is far less than 2. For other cases in which the release rate is doubled the value of $(T_{U_2}/T_{U_1})^4$ will be somewhat different, and will be increased by using a furnace with a larger fraction cold and also by using a higher temperature of the combustion air; this value will become nearly equal to the ratio of energy release rates when the fraction cold is 100 per cent and the air temperature is 500 deg. fahr. The steam flow rate is not exactly proportional to the energy release rate, but for the purpose of this discussion the variation from proportionality is not serious.

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[9] See Proc. N.E.L.A., 1930, p. 1066.

[10] Power Plant Engineering, Feb. 1, 1932, p. 128.

Reliable data on heat transmission in superheaters are not easily found, but some additional references [11] are here given.

489. Reheaters.—(a) The reheating of steam [12] in order to take advantage of using high-pressure steam in turbines without exceeding the maximum limit of moisture in the exhaust, is usually effected by means of *hot gases, steam,* or *both.* The reheaters of the usual systems will be considered first; later some unusual ones will be discussed.

(b) **Gas reheaters** are generally located in steam-generating units, either in the same positions as are superheaters, or else in the exit-gas passages; and they may be of either the **convection type** or the **radiant** type. In effect they are superheaters, but they handle much larger volumes of steam than do the usual superheaters because the density of the fluid flowing through them is much lower. An example of a convection type of reheater is given in Fig. 805, which shows the reheater that is a part of the unit illustrated in Fig. 735, and for which the estimated performance data are given in Table LXVI, p. 690. Other examples of reheaters of this type are given in figures which follow. Reheaters of the radiant type resemble radiant superheaters, examples of which were given in the preceding section.[13] Steam-generating units containing reheaters are often designated as *reheat boilers* or *reheat units*; in some, as in Fig. 806, only enough boiler-tube surface is provided to screen the reheater coils and reduce the furnace gases to 1100 or 1200 deg. fahr., the main supply of steam for the turbine coming from one or more standard steam generators. Other reheat units, in addition to serving as reheaters, also furnish the full steam requirements of the turbine served by them, and in some cases only one such unit is used with each turbine. Steam-generating reheat units of this kind are shown in Figs. 735 and 807. In the latter the reheater occupies the position usual for an interpass superheater. The principal disadvantage of using gas reheaters lies in the long pipe of large diameter needed for conducting

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[11] "Heat Transfer in Superheater Element," by John T. McIntyre, Trans. Inst. of Marine Eng., Vol. 39, 1927–28, p. 590. "Design of Superheaters," by R. M. Gates and C. W. Gordon, Power, May 28, 1929, p. 869; also discussion by F. Michel, Power, Aug. 6, 1929, p. 220. "Analysis of Heat Absorption in Boilers and Superheaters," by N. Artsay, Trans. A.S.M.E., 1929, FSP–51–37, p. 247. "Heat Transfer in the Locomotive Superheater," by L. H. Fry, Trans. A.S.M.E., 1930, RR–52–3, p. 17. Proc. N.E.L.A.: 1928, p. 1170; 1929, pp. 1203, 1284; 1930, p. 1067; 1931, p. 756. "Radiant-Superheater Development," by M. K. Drewry, Trans. A.S.M.E., 1932, FSP–54–14, p. 181. "Regulation of Steam Temperature by Controlled Gas Flow," by E. V. Rieder, Combustion, Nov. 1934, p. 9.

[12] For a general discussion of the application of reheat, the best reheat pressures, and results obtainable, see Sect. **276**, p. 117. For the definition of reheating surface and its measurement according to the A.S.M.E. Code, see p. 437.

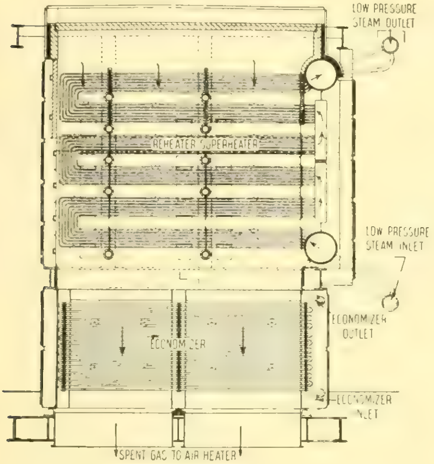

FIG. 805.—Convection Type of Reheater.

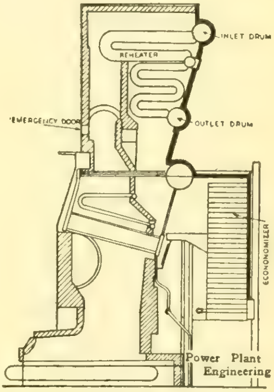

FIG. 806.—Early Reheat Boiler. (Crawford Ave.)

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[13] For experiences with a screened radiant superheater see, "High Pressure Boiler and Turbine Operation at Northeast Station," by J. A. Keeth, Combustion, Oct., 1931, p. 31.

the steam from the turbine to the reheater and back to the turbine again.[14]

(c) **Steam reheaters** usually consist of cylindrical shells in which the low-pressure steam is heated by being passed over tubing through which flows high-pressure saturated steam tapped from the boiler drum or superheated steam obtained from the pipe line. In either case, with a single reheater the reheat temperature is limited to a value near the saturation temperature of the high-pressure steam. With high boiler pressure this limitation is less objectionable than with low pressure. The amount of live steam used for reheating is relatively small compared with the quantity heated, and the efficiency of heat transfer is very high.

If a steam reheater is placed close to the turbine served, the reheat piping is short, and the pressure drop through the reheat system is small; if ample surface is provided, the control of the temperature is automatic and independent of the load, and no special regulating devices are required on the turbine; but pumps or other means for returning the condensate from the heater to the boiler or to the hotwell are required.

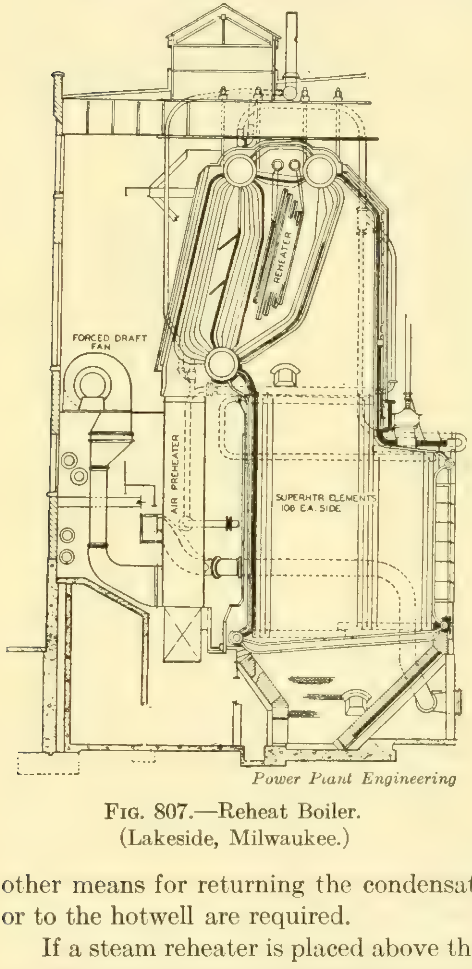

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FIG. 807.—Reheat Boiler. (Lakeside, Milwaukee.)

If a steam reheater is placed above the boiler, as in Figs. 539 (p. 434) and 735 (p. 688), the high-pressure drip therefrom can be drained direct to the latter, but long pipe lines are required for transmitting the steam between the turbine and the reheater.

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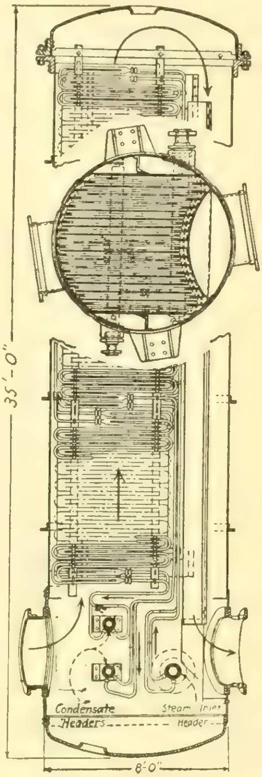
[14] The piping to the reheater shown in Fig. 806 is 30 in. in diameter and about 100 ft. long each way. Careful provision must be made with such piping for its proper support, involving due allowance for its expansion and contraction with changes in temperature and pressure. For a description of this unit, see Proc. N.E.L.A., 1926, p. 1200; for tests of a similar unit, see Proc. N.E.L.A., 1927, p. 914.

One arrangement of a steam reheater which is located near the turbine is shown in Fig. 808; in this design the counterflow principle is used and zigzag tubing is so proportioned that the velocity of the heating steam within it is high while this steam still has superheat and is low after condensation begins. The tubes in other designs are often in the form of simple U-bends, as in Fig. 810, or are arranged in coils.



Power Plant Engineering

FIG. 808.

Reheat temperatures considerably above the saturation temperature of the high-pressure steam can be obtained by using two steam reheaters in series and applying the counterflow principle. The second heater is of the *dead-end type*, so called because none of the high-pressure fluid flows out of it except as condensate, which is delivered to the boiler feed system.[15]

(d) **Reheater control problems** involve (1) maintaining constant reheat temperature at all loads, (2) protecting the reheater elements, the piping, and the turbine from exposure to excessive temperatures in emergencies, and (3) avoiding the possibility of overspeeding the turbine.

Reheat temperature regulation may be effected in the following ways: (1) If there is a separately fired reheater, the rate of firing can be varied to accomplish the purpose. (2) With convection reheaters, a movable gas-baffle can be used. (3) A reheater may be selected with a temperature characteristic opposed to that of the prime mover supplying the steam to be reheated. (4) The regulation may be accomplished by using a steam reheater in advance of a gas reheater and controlling manually or thermostatically the influence of the steam reheater, to which the extraction steam goes first. (5) Desuperheaters may be used.

Illustrations of case (3) are found in those central stations which have superimposed a new high-pressure system (say 1200 lb.) on an old low-pressure one (say 300 lb.). The exhaust from the high-pressure turbine at all loads is delivered to the reheater with a constant pressure which is the same as that in the steam main connected to the low-pressure superheater. If the load in the high-pressure turbine is low, the governor will throttle the steam admitted, and, since the exhaust pressure is constant, the temperature of this exhaust will be higher than that at full load. Consequently, a *convection type* of reheater is needed to maintain substantially constant temperature of this reheated steam at all loads. On the other hand, when the steam to be reheated is taken from a compound turbine, its pressure and temperature will both drop as the load falls off; hence, for this case, a *radiant type* of reheater is required to give approximately constant temperature of the reheated steam.

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[15] See Power Plant Engineering, July 1, 1931, p. 697.

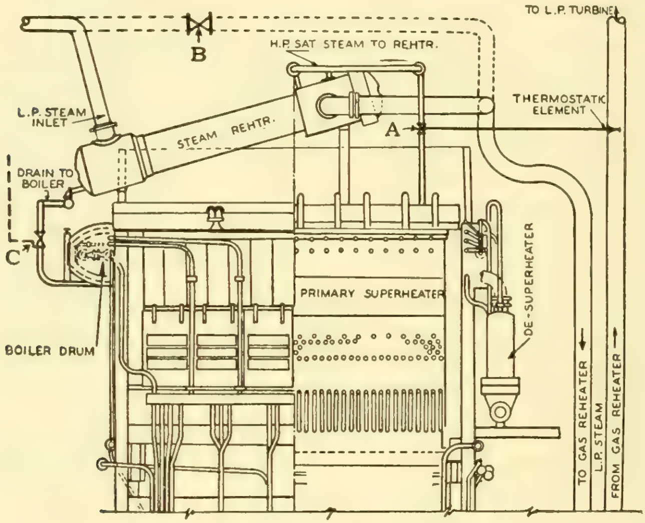

Fig. 809.

With a combination of steam and gas reheaters the control of temperature may be effected by providing means for (*a*) regulating the flow of the high-pressure heating steam to the reheater, (*b*) by-passing a proper proportion of the low-pressure steam around the steam reheater, or (*c*) controlling the level of the condensate in the steam reheater. These types of control are shown at *A*, *B*, and *C* in Fig. 809, in which the arrangement of the steam reheater is much like that in Fig. 735.

When two or more reheat boilers deliver steam to the same steam line, control devices must be added to distribute the flow of steam between them in proportion to their rates of firing.

Protection of gas reheater coils against excessive temperatures when the load drops suddenly can be accomplished by (1) opening emergency doors in the walls of the setting (see Fig. 806), either by hand or by automatic means, or (2) by-passing some live steam direct to the gas heater. Such protection of the reheat coils also automatically protects valves, piping, and other equipment in the reheat system from excessive temperatures.

Prevention of overspeeding of the turbine is of the utmost importance. With the cross-compound arrangement of turbines, speed-control governors are usually provided on each element of the unit, and excessive crossover pressure is avoided in emergencies by the operation of a relief valve located in the return pipe from the gas reheater. Further equipment is required to prevent the low-pressure turbine from being overspeeded by the steam which is not under governor control in the reheater and piping. This may consist of an interceptor valve which is placed in the extraction piping close to the low-pressure turbine, and is controlled by an overspeed governor.[16] Usually the steam thus by-passed goes to the condenser wherein diffusers are used to distribute it over a large surface.[17]

(e) **Some proposed reheat systems** include the substitution of a more suitable fluid for high-pressure steam used in indirect reheaters, and the employment of progressive reheating by steam bled from the turbine.

The **use of diphenyl** in tubular reheaters has been suggested by Cunningham.[18] As this material has a critical temperature of 980 deg. fahr. and pressure of 592.3 lb./sq. in., it has relatively high saturation temperatures at moderate pressures. This material can be heated in a special unit, or in elements included in the steam-generating setting, and can be circulated through the reheater in much the same manner as is done with diphenyl oxide in the air heater system shown in Fig. 736. **Mercury** and **diphenyl oxide** are other substances that may be used as reheating fluids.[19]

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[16] For description of one of these arrangements, see Power, Jan. 12, 1926, p. 54.

[17] Additional references: Series of articles on Reheat Practice, Operation, and Control, Power Plant Engineering, July 1, 1931, p. 697; Aug. 15, 1931, p. 840; May 1, 1932, p. 370; June 15, 1932, p. 484; Aug. 1, 1932, p. 588.

[18] "Diphenyl May Solve the Reheater Problem," Power, Sept. 2, 1930, p. 374. (This article also contains a table of the physical properties of diphenyl.)

[19] Note that there is a distinction between diphenyl and diphenyl oxide. The properties of the latter are discussed briefly in Part I, p. 365, also in Mech. Eng., Aug., 1926, p. 815.

In a **regenerative reheating system,** suggested by DeSmaele,[20] the steam passing from the high-pressure turbine element to the low-pressure one flows through a reheater containing a series of independent tubular elements that are supplied with *steam bled from different stages* of the high-pressure turbine; these elements are arranged in the order of increasing temperatures, and thus the low-pressure steam is heated progressively. These elements may also be supplemented by a final one which is heated by live steam. The bleeding points can be the same as those used for regenerative feedwater heating, and the corresponding steam-heating and water-heating elements can be arranged in series so that the steam from one bleeding point first passes through a reheater coil and then goes to the corresponding feedwater heater where it is finally condensed. With such a system from 25 to 30 per cent of the steam is bled for reheating, and consequently less condenser surface is required. Compared with gas reheating, this arrangement has the advantage that all its apparatus is in close juxtaposition to the turbine; and compared with the usual steam reheater, a higher reheat temperature results provided live steam is used in the last coil.

The feasibility of using any reheating system depends not only upon the performance characteristics, but also on the economies effected in first cost and operation, as compared with those of the more usual systems which it might replace.

490. Desuperheaters.—(a) Some of the purposes for which desuperheaters are used are the following: (1) to control accurately the temperature of the superheated steam used in prime movers, or in industrial or chemical processes; (2) to permit steam from a modern section of a plant to be used in an emergency, in combination with pressure reduction, in old equipment which was not designed for high pressures and temperatures; (3) to provide low-temperature steam to replace that normally taken from the turbine exhaust or extraction point for use in a low-pressure, low-temperature system; and (4) to provide low-temperature or saturated steam for industrial processes requiring steam in such small amounts as not to justify installing and operating a special boiler in addition to the large steam generators in the same plant.

Desuperheaters may be divided into **two general types:** (1) the *noncontact* or *surface type*, and (2) the *direct-contact type.*

(**b**) The **surface type of desuperheater** resembles a closed feedwater heater, as it consists of a shell within which are tubes which separate the steam from the water; the steam usually flows through the tubes and the water is on the outside of them; Fig. 810 shows the arrangement of a large desuperheater of this type.[21] Such desuperheaters assure against having slugs of water in the steam; and they are not of prohibitive size, because of the high heat transfer rates obtained in them by using high velocity of steam through their tubes.[22] The water which is used to absorb the superheat is generally delivered to the boiler feed; that part of it which has become steam goes to the steam drum of the boiler. The temperature is regulated either by varying the height of the water in the apparatus or by by-passing some of the steam.

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[20] Thesis, "The Utilization of High Pressure and High Temperature Steam for the Production of Power," Cornell University, June, 1930.

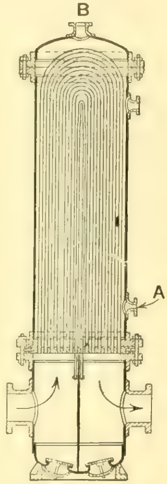

FIG. 810.—Large Surface Type of Desuperheater.

(c) In the **direct-contact type** of desuperheater a sufficient amount of water is *added to the steam* to reduce its temperature the right amount by being itself converted into steam which is added to that which has been desuperheated. In the **cartridge type** of direct-contact apparatus the steam is made to flow through water-soaked pads or cartridges of woven wire or similar material. In the **spray type,** one example of which is given in Fig. 811, one or more jets of water are sprayed into the flowing steam and a steam separator is used to remove any surplus water. In the **carburetor type,** the steam passes through a restricted throat wherein the water is introduced in suitable amounts through a small nozzle and becomes instantly entrained in the steam in the form of vapor.

In all the foregoing direct-contact devices the water supply may be controlled either by hand or by automatic thermostatic regulators. The difference in pressure established by the flow of steam through a restricted throat may be utilized to control the flow of water from a constant-level reservoir, as in the Smoot desuperheater, with the result that the ratio between the steam flow and water flow remains constant since the same pressure difference controls the flow of both fluids.

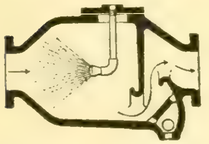

FIG. 811.—Spray Type of Desuperheater.

(d) The **water used** in desuperheaters should be of a purity suitable for the type of apparatus used. Any water satisfactory for boiler feed may be used with the *surface type*, but the concentration of salts therein must be controlled, as in boiler operation, by periodic or continuous blowdown. With the *direct-contact type* any soluble salts in the water used will be carried along in the steam as solids and may clog strainers and cause valves to stick. Condensate is suitable for use with desuperheaters of this kind.[23]

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[21] Described in "New Field for Desuperheaters," by B. N. Broido, Power, Nov. 30, 1926, p. 803.

[22] "High Heat-Transfer Rates for Surface Type Desuperheaters," by P. H. Hardie, Power, Feb. 15, 1927, p. 253.

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[23] "Importance of Water Supply for Desuperheaters," by C. E. Butler, Jr., in Power Plant Engineering, Sept. 15, 1930, p. 1056.

ADDITIONAL REFERENCES

"Radiant-Superheater Developments," by M. K. Drewry, Trans. A.S.M.E., 1932, FSP–54–14, p. 181.

"High-Temperature Steam Experience at Detroit," by P. W. Thompson and R. M. Van Duzer, Jr., Trans. A.S.M.E. 1934, FSP–56–9, p. 497.

"Superheat Is Responsible for Higher Efficiencies," Power Pl. Eng., Jan. 1934, p. 19.

"Regulation of Steam Temperature by Controlled Gas Flow," Part I, by E. V. Rieder, Combustion, Nov. 1934, p. 9; Part II, by C. W. Gordon, Dec. 1934, p. 10.

CHAPTER XLI

ECONOMIZERS AND AIR HEATERS

491. Introduction.—(a) The recovery of heat from the gases leaving the boiler of a steam-generating unit may be effected by the use of either **economizers,** or **air heaters,** or both in combination. The function of the former is to transfer heat from the gases to the feedwater; that of the latter is to transfer heat from the gases to air which is to be used for combustion. In modern practice these devices are designed as integral parts of the steam-generating unit, especially in the sense that they must justify their inclusion from an economic standpoint; hence the amount of heat recovery assigned to either or both, in a given installation, depends on the relation which each bears to the plant as a whole. The extent to which economizer surface should be installed depends primarily upon the temperature to which the feedwater has been raised in other parts of the plant, such as in extraction heaters, before reaching the economizer; the amount of surface in the air preheater is governed by the economical limit of temperature (and also of volume) for the combustion air. When the two pieces of equipment are used in combination, an economic apportionment of heating surface between the boiler, economizer, and air heater must be made, as explained in Sect. **482** (i), and a study of each individual case is necessary in order that the proper combinations of proportional heat recovery and of exit gas temperature may be determined.

(b) The **primary advantages** resulting from the recovery of heat by economizers and air heaters include (1) the thermal gain, with a resultant elevation of the efficiency curve and a saving of fuel; (2) the flattening of the efficiency curve in the region of high forcing, or the ability of the boilers and furnaces to operate at greater capacity while maintaining high efficiency. This last may, in turn, permit the installation of smaller furnaces and boilers, or the use of fewer units of the same size, for a given output, thus permitting a reduction in the initial expenditure for the boiler-room equipment and the building. However, from an economic standpoint, the installation of heat-recovery apparatus is justified only when the cost of fuel is relatively high and the units are rather heavily loaded most of the time; otherwise the saving in fuel will not pay for the charges against the purchase and operation of the additional equipment.

(**c**) Heat-recovery equipment also possesses **certain disadvantages,** those common to all forms being as follows: (1) Such apparatus not only impedes the flow of the products of combustion through the gas passages of the system, but it also lowers the stack temperatures on which the production of natural draft depends. This usually necessitates the installation of **mechanical draft equipment,** and cognizance must always be taken of the initial cost of this extra apparatus, of the space it occupies, and of the operating expense for its power consumption, attendance, and maintenance. (2) **Serious corrosion** will occur, not only in the heat-recovery equipment itself, but also in any unprotected metallic flues and stacks into which the gases flow, if the products of combustion are cooled below the dew point of the sulphuric acid vapor in them. Either the gas temperature should be maintained sufficiently high to avoid this difficulty, say above 240 deg. fahr.,[1] or the parts affected should be protected or be of non-corrodible material. (3) The **extra cost** involved for the equipment, and for its maintenance and replacement, should not be overlooked.

492. General Characteristics of Economizers.—(**a**) When an economizer [2] is used for heating the feedwater, which is the only application considered here, the amount of heat recovery is such that, for moderate rates of operation, each 10 or 11 deg. rise in the temperature of the feedwater, or each 40 deg. decrease in gas temperature, reduces the total fuel cost about 1 per cent, provided there is no air infiltration or radiation loss associated with the apparatus. In many cases where all the feedwater heating is done by the economizer, the saving in fuel, or the increase in steam-generating capacity, may amount to 10 or 15 per cent, or even more; when the economizer is used to supplement regenerative feedwater heating, the gain due to the economizer alone is from 3 to 7 per cent. In the latter case the increase in economy brought about by the use of the combination is greater than that which would result from either method of feedwater heating alone, while, with either arrangement, a further gain in economy may result because of the improved condition under which the steam-generating unit operates. The effect of using an economizer in a steam-generating unit which, even without it, has a very high efficiency, is shown in Fig. 812; test results from other units are given in Fig. 544 (p. 452), and in Table XLVII (p. 454).

(**b**) In addition to the primary advantages possessed by all heat-recovery apparatus, as given in Sect. **491** (**b**), economizers have the following **secondary advantages:** (1) They provide a reserve supply of hot water to help the unit meet fluctuating or sudden heavy demands for steam, (2) they protect the boiler from receiving cold feedwater, and (3) they make possible the precipitation of some of the impurities in the feedwater which might otherwise form scale in the boiler.

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[1] The exact value depends on the partial pressure which the vapor may have.

[2] For the definition of economizer surface and its measurement in accordance with the A.S.M.E. Code, see p. 437.

(c) The **important requirements** which economizers must meet are as follows: (1) The *exit gas* should not be cooled to the *dew point* of the sulphuric acid vapor contained in it. (2) The *entering water* should be at a temperature high enough to prevent its occluded or dissolved gases, of which oxygen is the most active, from causing internal corrosion of the tubes; deaerated water should be above 120 deg. fahr., and water not deaerated should be above 180 deg. fahr. (3) The *tube walls* should be at a temperature sufficiently high (at least 165 deg. fahr.) to prevent external sweating which would cause the adhering soot to become pasty and difficult to remove. (4) The *feedwater* should not be heated to its boiling temperature by the economizer unless the tubes and headers are designed to handle the very much larger volume possessed by the outgoing fluid when steam is formed; usually a temperature margin of 25 to 50 deg. is allowed, the amount depending on the widest pressure variation which may occur within the boiler. (5) The *resistance to gas flow* through the passages should be as low as possible. (6) The *size* and *shape* of the apparatus should conform to the space limitations, and the arrangement, of the entire unit. (7) Provision must be made for *inspection* and *repair*, and for internal and external *cleaning.* (8) If the apparatus is provided with inlet and outlet by-pass valves, it must also be equipped with *relief valves.* (9) *Blowoff valves* must be provided in some cases. (10) Provision should be made against the loss of heat by *radiation* and against the *infiltration of air.* (11) The equipment should be installed so as to operate on the *counterflow principle* with the gases passing downward and the water upward, although a complete arrangement of this kind is not possible in some cases.

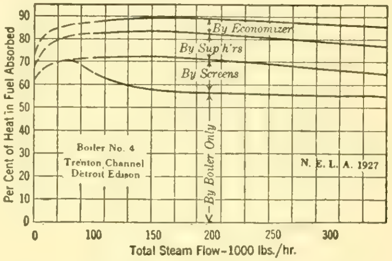

FIG. 812.—Results of Tests of a Steam-Generating Unit, Showing Effect of Economizer.[3]

(d) The **heating surface** of economizers is made of **cast-iron** or **steel tubes** which may be either straight or curved, and which are located between headers, junction boxes, or drums, so that a definite circulation can be maintained. As a tube material, cast iron resists corrosion better than does mild steel, but it is not so suitable for use with high pressures, the limit assigned by different engineers varying from 150 to 400 lb./sq. in. By arranging the pumps in series with the economizer between, the pressure in the latter may be maintained at a value much below that in the boiler, thus permitting the use of cast-iron construction with high-pressure boilers; however, steel-tube economizers are almost universally used in the high-pressure field, with the ordinary arrangement of pumps. Tubes are occasionally made of non-corrodible material, and if composed of cast iron or tube steel, may be covered with protective coatings, such as lead, paint, or enamel. In some designs the tubes are encased in protecting sleeves which also provide extended (gilled) surfaces.

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[3] For boiler No. 4, Trenton Channel plant of the Detroit Edison Co., see Proc. N.E.L.A. (1927), p. 932.

493. Types and Arrangements of Economizers.—(a) The **first economizers** were of the **detached type** and were really added equipment which was used to supplement boilers of the customary proportions. Such economizers are still used extensively, especially in small plants, and are often arranged so that one economizer will handle the exit gases from, and the feedwater to, several steam-generating units. In this case the apparatus is designated as a **group-type** economizer and is usually provided with by-passes for the gas and water so that all the units connected with it may continue in operation even though the economizer is out of order.

The **special advantages of the group type** of economizer are (1) simplicity, (2) relative cheapness, (3) the small amount of attention required, and (4) the possibility of installation in any convenient location near the units served, or even out of doors in order to reduce the size of the building.

The **disadvantages of the group type** are (1) the reduced effectiveness because of air infiltration through the walls of the long flues between the boiler and the economizer, and also through the by-pass dampers and through any units not in operation; (2) the loss of heat through the extensive flues and economizer walls; (3) the effect on all the units of the group if the economizer is out of order; (4) the difficulty of providing suitable space for such large equipment in some cases; and (5) the large draft losses through the apparatus.

(b) In **modern practice,** economizers of the individual or **unit type** are the ones generally used in connection with large boilers. Each economizer is designed for installation with a single steam-generating unit, and is sometimes constructed as a part of the boiler itself, in which case it is of the **integral type,**[4] and occupies a minimum of space, often serving to replace some of the boiler heating surface which would otherwise be required. Unit economizers possess all the advantages of the group type, except possibly that of low cost, and avoid or minimize the disadvantages of the latter.

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[4] Such an apparatus is sometimes referred to as the *economizer element,* the *water preheater,* or the *preboiler element.*

The usual economizer is intended solely as a feedwater heater; but sometimes the economizer may be arranged to generate a certain amount of steam, in which case it is termed a **steaming economizer.** The quantity of steam so generated may be as much as 20 per cent of the total amount produced by the unit when forced.

The various features and arrangements of the different types of economizers can be studied best by referring to illustrations of actual practice.

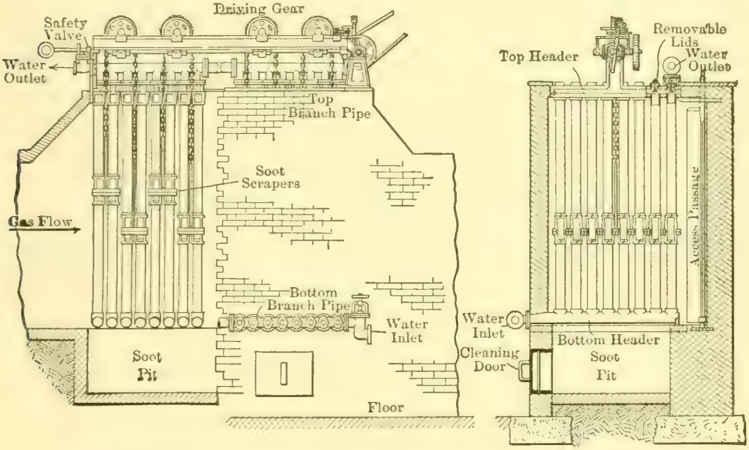

FIG. 813.—An Early Form of Green Economizer.

(c) The **Green Economizer,** shown in Fig. 813, exemplifies the **detached type.** It is composed of a series of sections, each having vertical cast-iron tubes (with an outside diameter of $4\frac{9}{16}$ in. and a usual length of 9 to 12 ft.), which are connected by means of forced fits to transverse cast-iron headers. The length, width, number, and grouping of sections forming this economizer can be varied to suit the conditions encountered in each case. The sections are connected at top and bottom, on opposite sides, by longitudinal branch pipes which are attached to the upper and lower headers respectively, and are located outside of the casing. Power-driven scrapers constantly move up and down over the external surfaces of the tubes to remove the soot deposits which then fall to the pit below. The upper headers and lower branch connections contain handholes, the covers of which can be removed to provide for internal inspection and cleaning. The water is preferably introduced into the lower branch pipe at a point farthest from the boiler, and is discharged from the upper branch at the nearest point to the boiler. The setting is either of brick or of sheet-steel panels lined with non-conducting material. Fig. 814 shows a simple group arrangement which permits the gases to be by-passed direct to the stack when the economizer is not in use, the feedwater also being by-passed direct to the boiler. The power required for driving the scrapers is from ½ to 1 hp./1000 sq. ft. of economizer surface.

Other detached economizers, such as that shown in Fig. 539 (p. 434), use steel tubes and may be constructed in a manner similar to some of the types illustrated later in this section.

(d) An **integral economizer** of simple and cheap form is shown in Fig. 815. In this arrangement the feedwater is introduced into the bottom drum D and flows upward through the tubes in the economizer element E, in a direction opposite to the gas flow. A somewhat similar arrangement is shown in Fig. 748 (p. 698).

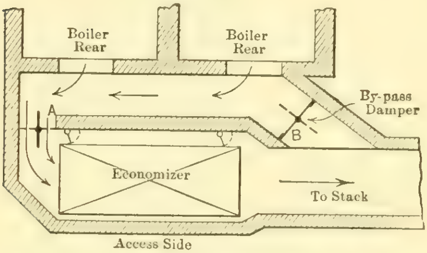

FIG. 814.—Group Arrangement of Green Economizer.

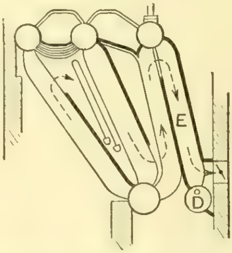

FIG. 815.—Integral Economizer or Water Preheater.

Other examples of integral economizers are given in Fig. 734 (p. 687) and Fig. 740 (p. 693); in these arrangements the economizer is substituted for the second and third passes of the conventional boiler. These economizers use the counterflow principle, and therefore may be cheaper and may occupy less space than the boiler surface they replace.

(e) An economizer of the **steaming type** is shown in Fig. 734 referred to above. The multiple connections shown between the economizer and the steam drum of the boiler are necessary in order that adequate passage area will be provided for the discharge of the steam generated in the economizer. Provision must also be made, by means of an intermediate connection, for maintaining a safe water level in the boiler at all times. As there are no valves between such an economizer and the boiler, relief valves are not required on the former.

(f) **A two-drum bent-tube economizer** of the unit type, and of comparatively simple and cheap construction, is illustrated in Fig. 538 (p. 432). It is located within the main setting of the steam-generating unit, and the partition in the upper drum causes the water to flow down through the rear bank of tubes and return upward through the front bank in directions countercurrent to the gas flow.[5]

(g) **A multiple-return-bend economizer** is shown in Fig. 805 (p. 755); it constitutes a part of the 1450-lb. B. & W. unit illustrated in Fig. 735 (p. 688). The 2-in. continuous return tubes, the great length of which is made possible by fusion welding, are connected in parallel to inlet and outlet headers and are staggered vertically. The hot water from the economizer is delivered by a single pipe to the steam drum of the boiler. Data regarding this economizer and its expected performance are contained in Table LXVI, p. 690.

A slightly different design of B. & W. economizer has a heating surface made up of a series of continuous hairpin loops, which are joined at the ends by means of detachable solid-forged return bends, as shown in Fig. 816. The first loop of each series is joined to the inlet header by a similar return bend. The tubes are staggered vertically, and the return bends and inlet header are located outside of the path of the gases so that their joints are protected, and are readily accessible while the economizer is in service. The interior of the tubes, and of the inlet header, can be reached by taking off the proper return bends; hence no handholes are required in this design. The connection to the boiler drum from the top row of economizer tubes may be made by carrying the tubes up to the drum, as shown in Fig. 734 (p. 687), in which case no intervening collecting header is necessary.

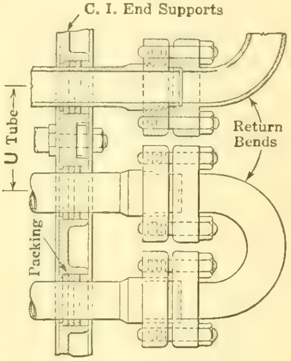

Fig. 816.—B. & W. Return Bends.

(h) **Extended-surface economizers** give a compact arrangement which minimizes the space required for their installation. Such economizers are exemplified by the Combustion Engineering, Foster Wheeler, and Sturtevant designs, which are described below.

The **C–E economizer,** shown in Fig. 817, is made up of U-tubes having longitudinal fins like those on the fin tubes of water walls. The open end of one tube is connected to the next tube in series by means of return bends and flanges which are located outside of the cast-iron tube sheets and are thus out of the gas passage as well as accessible at all times. Usually there are two separate water circuits extending in parallel between common inlet and outlet headers. An installation of one of these economizers is shown in Fig. 575 (p. 492).

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[5] Performance data for a somewhat similar economizer, but with different baffling, are given in Proc. N.E.L.A., 1927, p. 911.

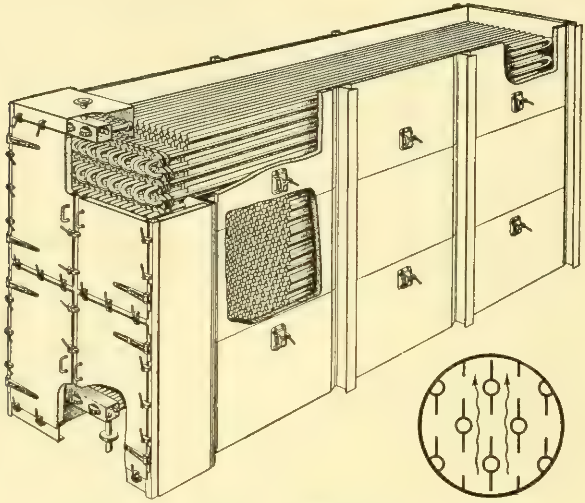

FIG. 817.—Combustion Engineering Economizer.

The **Foster Wheeler** extended surface economizer is composed of horizontal elements arranged as shown in Fig. 818. These elements consist of 2-in. steel tubes encased in cast-iron protecting sleeves with gilled rings, the tube ends being expanded into forged-steel return headers. Handholes are provided in the headers for inspecting and cleaning the tubes; and the construction is such that any element can be removed without disturbing the others. Because the cast-iron rings entirely cover the tubes, the water may be drained from the economizer

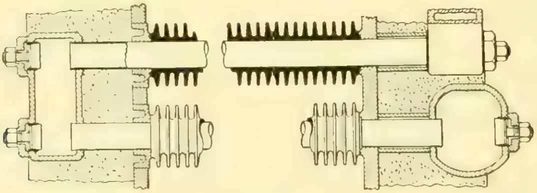

FIG. 818.—Elements of Foster Wheeler Economizer.

and any joint may then be re-rolled without shutting down the boiler. The installation shown in Fig. 819 uses the counterflow principle and is equipped with soot blowers for cleaning the elements while in operation; the surfaces may also be washed with water when the unit is out of service.

The **Sturtevant economizer,** which is also of the extended surface type, is built up of sections consisting of horizontal steel tubes located one above the other, and joining forged- or cast-steel front and rear headers, into which they are expanded. Steel discs are applied to the tubes to increase the heating surface, after which each tube element is lead coated to protect the surfaces from corrosion. The front header of each section has a partition across the center, and the water enters the lower half of this header, flows to the rear header, reverses, then returns to the upper half of the front header, and finally flows through a return bend into the lower half of the next front header above. The method of support is such that any section can be disconnected from those above and below it, and withdrawn, and a new one inserted in the gap, without disturbing the adjacent sections.

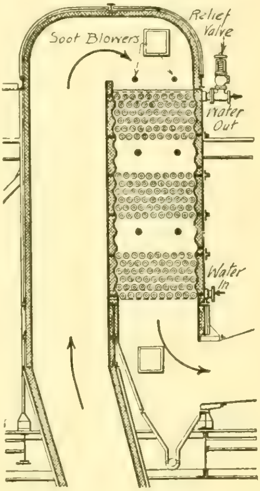

FIG. 819.—Installation of Foster Wheeler Economizer.

494. Air Heaters.—(**a**) In addition to the primary advantages resulting from the use of any form of heat-recovery equipment, as stated in Sect. **491** (**b**), the following **additional advantages** are derived from the use of air heaters[6] (also called preheaters): Through the use of heated air, (1) complete combustion is more nearly attained in the furnace, (2) less excess air is required, (3) the furnace temperature is higher, and (4) a greater proportion of the energy released in the furnace is transmitted to the heating surfaces by radiation, and less by convection; hence, (5) the furnace and boiler efficiencies are increased, (6) the gas temperature at the boiler outlet is lower with a given extent of boiler heating surface, or less surface is required to reduce the gas temperature at the boiler outlet to an assigned value, and (7) the heat-recovery apparatus is itself smaller than it would be if heated air were not used for combustion. Because of factors (1) and (2), the temperature of the furnace often increases more than the temperature of the combustion air in the preheater. Roughly, a decrease of about 33 deg. in the temperature of the exit gases should effect a saving of 1 per cent in fuel consumption, provided the reduction in gas temperature is due only to the absorption of heat by the preheater.

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[6] For a definition of air-heater surface and its measurement in accordance with the A.S.M.E. Code, see p. 437.

(**b**) When the fuel is **coal**, the use of heated combustion air has the following beneficial results: (1) the loss due to unconsumed carbon in the refuse is reduced; (2) ignition is more rapid, the flame is shorter, and less of the combustion process is carried into the spaces between the boiler tubes, thus minimizing tube slagging and tending to reduce the exit gas temperatures; (3) if the coal is moist, less space is required in the dryer, or in the furnace, for drying the coal prior to its combustion; (4) the combustion of relatively moist, poor, or low volatile coal is assisted, and cheaper grades may be burned; and (5) the draft loss through the boiler itself may be reduced, because of the decreased weight and temperature of the gas flowing, thereby partly offsetting the additional fan power necessitated by the presence of the preheater.

With **stoker firing,** the use of heated air may make the grate area more effective, and permit operation at higher rates of combustion per unit area of grate surface, especially with moist or smoky coals; also the rate of heat release under maximum forcing may be increased to possibly 80,000 B.t.u./hr./cu. ft. of furnace volume.

With **powdered coal, oil, or gas firing,** in furnaces having water-cooled walls, the use of heated air helps to maintain temperatures that will support combustion at lower rates of operation than are otherwise possible.

(**c**) In addition to the **disadvantages** listed in Sect. **491** (**c**) as applying to heat-recovery equipment in general, air heaters possess the following: (1) The ducts and stoker tuyères for handling heated air must be of larger cross-sectional area than those for cold air, or the velocities through them must be increased, since the specific volume of the air is often doubled by the heating. (2) The air heaters, ducts, and fans occupy a large amount of space. (3) The heat losses and air leakage through the ducts may be large. (4) The amount of heat recoverable in this way is limited by the minimum gas temperature that can be used without corrosion and the permissible maximum temperature to which the air may safely be raised, this figure being dependent upon the maintenance of proper operating conditions within the furnace and upon the allowable expansion of parts of the fuel-burning equipment.

(**d**) The **maximum air temperature** allowable with present designs of stokers and related equipment, even when these have been specially built for the conditions, is from 350 to 500 deg. fahr.; and although with powdered coal firing, the secondary air may be heated to a high temperature, some trouble has been experienced due to advance coking of the fuel when the primary air, which is mixed with the coal ahead of the burner, has a temperature exceeding 250 deg. fahr.

If the air is to be heated to a high temperature (e.g., 700 deg. fahr.), the gases must enter the preheater at a very high temperature; and in order that the size and cost of the preheater may be kept within reasonable limits in this case, the exit gases must leave this apparatus also at a relatively high temperature (e.g., 450 deg. fahr.). This results in a loss of energy which may not be justified by the financial saving due to the curtailment of boiler heating surface. For good overall economy, the temperature of the gases entering the preheater should probably be kept below 600 deg. fahr., and this result may often be obtained by placing a small economizer between the boiler and the preheater.

(**e**) The **benefits** derived from an air heater can be readily determined by test, when a unit is so arranged that it can be operated both with and without the preheater; but such arrangements are uncommon. In other cases, exact comparisons may be difficult to make. If the assumption is made that the unit has a constant fuel and air input, and that the combustion conditions and the temperature of the gases remain the same, then the heat returned from the preheater increases the quantity of steam generated, and the efficiency of the unit; and for a given output less fuel need be burned. However, in the actual case, the use of heated air may alter the combustion conditions considerably, and change the carbon, moisture, and hydrogen losses, the furnace temperature, the distribution of energy absorption by the various parts of the heating surface, and the gas exit temperature; hence there are also incidental gains which cannot be evaluated accurately.

495. Types and Arrangements of Air Heaters.—(**a**) The two general types of air preheaters are the **recuperative** and the **regenerative.** In the former type, heat is transmitted through metallic walls from the hot gases on one side to the air on the other side, as in **plate** and **tube preheaters,** the flow of gas and of air and the transfer of heat being *continuous* and *unidirectional.* In the latter type, the heat-transmitting surfaces, generally metallic plates, are exposed *alternately* to the heat-surrendering gases and to the heat-absorbing air. The recuperative heater is a fixed structure, whereas the regenerative heater has certain moving parts which are required to produce the necessary alternation of exposure.

(**b**) Air heaters of the **tubular type** consist of groups of tubes, either vertical, as in Fig. 820, or horizontal, which pass through tube sheets and are enclosed in a metal casing, generally of rectangular form. The hot gases usually flow through the tubes, around which the air makes one or more passes, preferably in a countercurrent arrangement. The construction should be leak proof and must provide for inequalities of expansion. Applications of tubular heaters are shown in Figs. 733, 734, and 735 (pp. 686, 687, and 688 respectively).

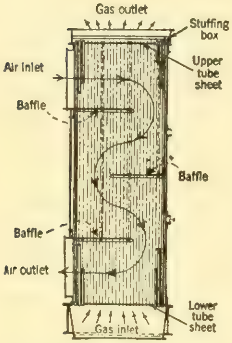

FIG. 820.—Tubular Air Heater.

(c) In **plate heaters** the gases and air flow in respective alternate narrow passages formed between closely spaced parallel plates. Guide strips are often used in these passages to direct the flow of both the air and the gas, so as to prevent stratification and eddying; and these strips also act as spacers and stiffeners. Fig. 821 illustrates one form of plate heater, as constructed by the Combustion Engineering Corp. The elements have sharp welded edges which offer little resistance to the gas flow; also, to eliminate leakage of air to the gas side, the entrance and exit portions of adjacent elements are welded together and to the casing, the connection to the latter being by means of sealing strips which hold the elements in place but also provide for expansion. Installations of C. E. air preheaters are shown in Figs. 746 and 751 (pp. 696 and 699).

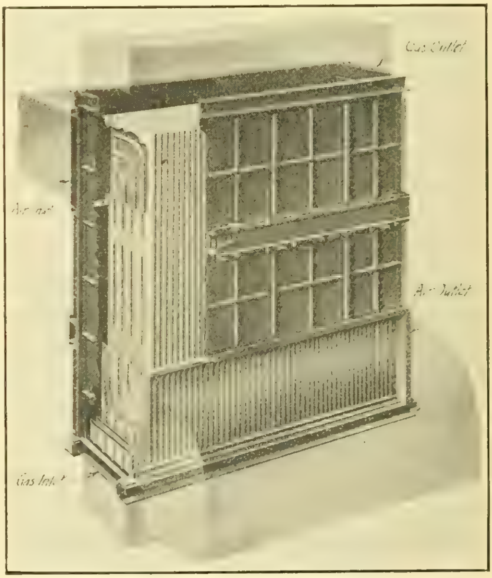

FIG. 821.—Plate Type of Air Heater.

In the Foster Wheeler preheater, which is of the plate type, the air makes several passes across the path of the gas, as in the installation shown in Fig. 539 (p. 434). In common with several other plate heaters, the edges of the plates composing the elements are welded together, forming a fixed arrangement.

As air heaters may be subject to corrosion at the end farthest from the boiler, several designs, such as the Sturtevant and Thermax, are provided with reversible elements so that the hot and cold ends may be interchanged, thereby prolonging the life of the plates.

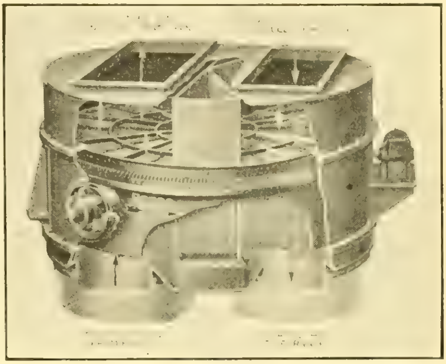

Fig. 822.—Ljungstrom Rotary Regenerative Air Preheater.

(d) In the **regenerative type** of air heater the elements are alternately heated by the flue gases and cooled by the combustion air. The **Ljungstrom rotary regenerative heater,** shown in Fig. 822, is so arranged that portions of the concentric rings of corrugated and flat plates, which form the slowly revolving rotor, are heated when in the stream of hot gases, and then move into the current of cold air to which the heat is surrendered. The gases are drawn through the left compartment by an induced-draft fan and the air is sent through the right side by a forced-draft fan. In the arrangement shown in the upper left part of Fig. 823, the two fans are mounted on the same shaft. Other arrangements of inlets and outlets and of fans are possible, and the axis of the rotor may be either vertical or horizontal. The claim is made that such heaters occupy much less space and have smaller weight than apparatus of the recuperative type, and that the power necessary to drive a preheater unit for a 10,000 sq. ft. boiler is less than $\frac{1}{4}$ hp. About 70 per cent heat recovery is

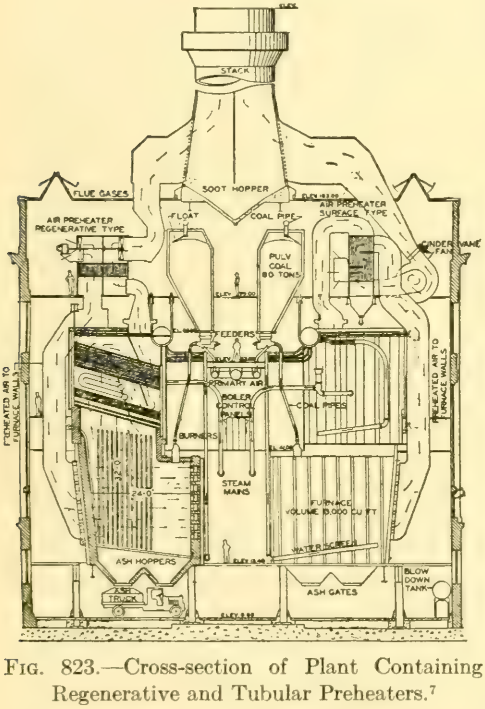

Fig. 823.—Cross-section of Plant Containing Regenerative and Tubular Preheaters.[7]

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[7] From "The Economics of Air Preheater Applications," by F. M. Van Deventer, Trans. A.S.M.E., 1927–28, FSP–50–16, p. 1236.

possible with commercially practical size, weight, and cost. The plant shown in cross-section in Fig. 823 contains preheaters of both the tubular and rotary-regenerative types above the respective boilers.

In another form of regenerative unit, the **Blaw-Knox,** the heating elements are stationary and are divided into groups through which cam-operated valves pass the gases and the combustion air alternately.

(**e**) All commercial types of air preheaters are arranged for easy replacement of parts, and for cleaning by means of air or steam lances, by permanently installed soot blowers, or by washing.

496. Special Applications of Economizers and Air Preheaters.—(**a**) In the recovery of heat from the gases leaving the boiler it is sometimes desirable to employ a preheater which raises the air temperature to a value too high for use in certain parts of the system. In this case the portion of the air which is required at a reduced temperature may be passed through an **air-to-water economizer.** If the air is to be supplied to stokers, it may be tempered in this way. When the primary air supplied to pulverized-coal burners must be at a temperature lower than that of the secondary air, the required quantity of the former can be diverted from the main supply and passed through such an economizer in order that its temperature may be reduced to the proper level, although in some cases the primary air is passed through a separate preheater provided for it alone. Hot air may also be reduced in temperature by passing it through an **air-to-air heater,** or by introducing cold air into it. Not only is an air-to-water economizer of value for tempering air, but when the gases handled are highly corrosive, this form may be desirable as a substitute for the usual gas-to-water apparatus.

A method which has recently been developed for the preheating of air makes use of the **indirect system,** whereby a liquid flowing in a closed circuit first absorbs heat from the gases leaving the boiler and later transfers this heat to the air. One example of the use of this method is described in Sect. **476** (**d**).

When the design of a steam-generating plant is such that preheated air must be available over a wide range of temperature, the desired result may be obtained by installing preheaters and economizers in parallel rather than in series, as is usually done. Such an arrangement is used at the Hell Gate station and is described in Sect. **397** (**d**).

(**b**) The **combination of** both an **economizer** and an **air preheater** into a single piece of apparatus is sometimes desirable. One such arrangement, called a **supermiser** [8] (British), is shown in Fig. 824. This apparatus is very compact, and the parts can be made readily accessible for cleaning.

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[8] See Combustion, Feb., 1930, p. 40.

(c) The recovery of **waste heat rejected from internal-combustion engines** using the four-stroke cycle[9] is often highly desirable since frequently over 50 per cent of the heating value of the fuel may thus be reclaimed. The heat obtained from the exhaust gases may be used for warming buildings, directly or indirectly, for heating water or air, and for other purposes. The hot jacket-water may be similarly utilized; sometimes it can be used directly. When the gases are used for heating water or generating steam, it is necessary to employ some form of **heat economizer,** which may resemble a simple cast-iron or steel boiler of either the water-tube or fire-tube type. So important has the heat recovery from internal-combustion engines become that it is now being applied in connection with central heating systems, motorships, industrial plants, and hotels.[10] The recovered heat may even be returned to the engine to perform additional work therein, as is done in the **Still engine,** which combines an internal-combustion engine and a steam engine in a single unit that generates its own steam, and in which the gases and steam act on the opposite sides of the same piston. The exhaust gases (at 900 deg. fahr.) from the combustion end of the cylinder are used to heat the cooling water which is then passed through the jacket of this end where it is generated into steam. This steam, at a pressure of 100 to 120 lb./sq. in., is used on the other side of the piston and is then condensed and used again as jacket water. The added power developed by the steam generated from recovered heat is from 20 to 30 per cent of that developed in the combustion cylinder, and a brake thermal efficiency of 40 per cent has been obtained from such a unit.[11] A supplementary oil-fired boiler may be provided for starting.

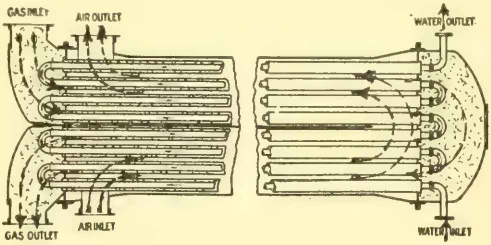

FIG. 824.—Section through a Supermiser.

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[9] The use of heat recovery equipment is inexpedient with two-stroke cycle engines, in most cases, because the dilution of the exhaust gases by the scavenging air makes the temperature head low and the apparatus very large.

[10] See Power Plant Engineering, Jan. 1, 1930, p. 68; also "Heat Recovery from Internal-Combustion Engines," by Clarkson and Bradford, Trans. A.S.M.E., OGP-53-4, 1931, p. 59.

[11] See Automotive Industries, June 19, 1919; "Still System of Internal-Combustion Engines for Marine Purposes," by F. L. Martineau, Inst. Mar. Engrs. Trans., v. 34, April, 1922, p. 37; also "Automobile and Aircraft Engines," by Judge, Isaac Pitman & Sons, Ltd. (1931), pp. 821–829.

(d) Many **applications** of apparatus for the recovery of waste heat, which will show good returns on the investment, may be made in various kinds of industrial plants; and the possibilities should always be carefully investigated when a supply of heat is available for use in this way. All investigations of this kind, as well as those for the usual types of economizers and air preheaters, always involve the laws of heat transmission for the case under consideration.

497. Heat Transmission in Economizers and Air Preheaters.—(a) In **economizers,** heat is transferred from the gases to the tubes by convection, through the tubes and deposits by conduction, and from the inside tube surface to the water by convection. The gas-film coefficient of heat transfer may be found from Eq. (420), pp. 271 and 272, when the mass velocity and temperature of the gas and the outside diameter of the tubes are known. The water-film coefficient of heat transfer may be estimated from Eq. (414), in which the velocity of the water is the most important governing variable. With the two film coefficients and the resistance of the metal path to heat flow known, the overall coefficient of heat transfer, U, may be found from Eq. (429 b). The gas film is the largest resistance to heat flow, and the mass velocity and temperature of the gas are therefore the principal variables governing the overall coefficient of heat transfer in the economizer. The results of the tests of four different economizers are plotted on logarithmic paper in Fig. 825. The temperatures of the gas entering and leaving the economizers were not given for A or B; but the arithmetic means of the entering and exit gas temperatures ranged from 343 to 388 deg. fahr. for economizer C and from 396 to 470 deg. fahr. for economizer D. For all these economizers, *the slope of the plot of U against the mass velocity of the gas on logarithmic paper is between 0.5 and 0.85*, and a straight line with a slope of 0.6 fits all the data fairly well. The conclusion is that, for these economizers, the overall coefficient of heat transmission, U, increases with the 0.6 power, approximately, of the mass velocity of the gas.

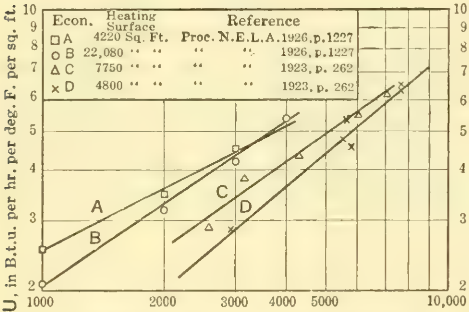


FIG. 825.—Test Values of the Overall Coefficient of Heat Transfer in Economizers.

The complete log of the test of economizer D is shown in Fig. 826;

this is a Foster steel tube economizer with a heating surface of 4800 sq. ft. installed with a Stirling boiler having a heating surface of 6865 sq. ft.

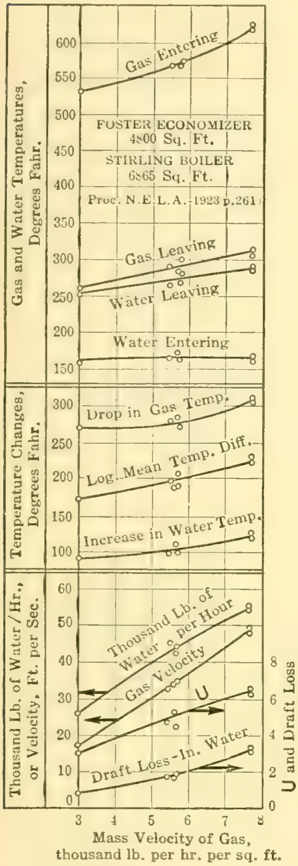


FIG. 826.—Complete Log of Economizer Test.

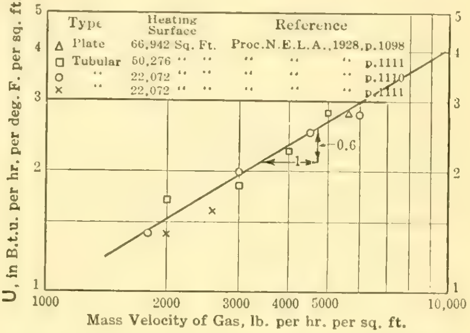


FIG. 827.—Test Values of the Overall Coefficient of Heat Transfer in Air Preheaters.

(b) In **air preheaters,** heat is transferred from the hot gases to the metal tubes or plates by convection, through the metal with deposits by conduction, and to the air by convection. The principal resistances to heat flow are the gas film and the air film; if the metal is reasonably clean, its resistance is practically negligible. In general, *the overall coefficient of heat transfer, U, is slightly lower, for the same mass velocity of the gas, in air preheaters than in economizers,* largely because the resistance of the air film to heat flow exceeds that of the water film in the economizer. The resistances of both films decrease as the mass velocity of fluid flow increases, and since the mass velocities of the air and gas increase simultaneously, U increases as either mass velocity increases. Since the film coefficients of heat transfer for both the gas and air increase about as the 0.6 power of mass velocity, as shown in Eq. (420) p. 271, U should also increase according to the same law. The results of tests on four different air preheaters are shown in Fig. 827 with U plotted on logarithmic paper against the mass velocity of the gas. A straight line with a slope of about 0.6 fits all these test points fairly well, showing that for these tests the overall coefficient of heat transfer increases about as the 0.6 power of the mass velocity of the gas. The mass velocity is such a predominating factor that, if *U* be divided by the mass velocity of the gas, in pounds per second per square foot, to the 0.6 power, the quotients for all the test values are nearly constant, lying between 2.0 and 2.5.

More complete data on the performance of a tubular air preheater are given in Fig. 828; and its overall coefficient, *U*, is also represented by the circles in Fig. 827. The coefficient of heat transfer, the mass velocity of the gas, and the pressure drops of the gas and air through the preheater are shown plotted against the rate of evaporation in Fig. 828. The drop in pressure of the air and gas in passing through the preheater increases with a greater power of the mass velocity or the rate of evaporation than does the overall coefficient of heat transfer.

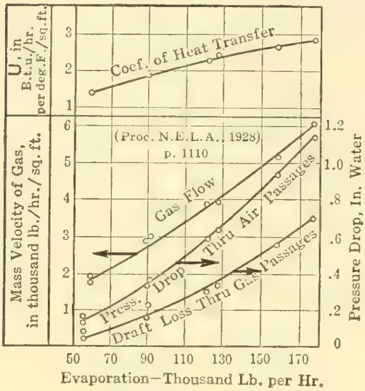


Fig. 828.—Results of Tests on Tubular Air Preheater with a Heating Surface of 22,072 Sq. Ft.

Additional References

"The Economics of Air-Preheater Applications," by F. M. Van Deventer, Trans. A.S.M.E., FSP-50-16 (1927), pp. 123–132; with discussion by Wohlenberg et al., pp. 132–147. "How to Check Your Economizer Guarantees," by B. M. Thornton, Power Plant Engineering, Feb. 1, 1929, pp. 171–173. "Dividing Heating Surface among Boiler, Economizer, and Air Heater," by L. Helander, Power, March 4, 1930, pp. 358–362. "Heat-Transmission Problems," by G. E. Tausley and O. Kubalek, Mechanical World and Engineering Record, Nov. 22, 1929, abstracted in Power, June 24, 1930, p. 999. "How Do You Select Your Economizer?" by T. Maynz, Power Plant Engineering, Nov. 1, 1930, pp. 1214–1219. "How to Check the Performance of Economizers," by B.J. Cross, Combustion, Dec., 1930, p. 48. "How to Check Air Heater Performance," by B. J. Cross, Combustion, Jan., 1931, pp. 50–51. "Plate-Type Air Heater Development and Present Status in the U. S.," by W. S. Patterson, Combustion, Nov. 1933, p. 7. "Heat Transfer in Air Heaters and Economizers," by H. Kreisinger, Combustion, Dec. 1933, p. 7. "Economizers and Air Heaters for Higher Efficiency," Power Pl. Eng., Jan. 1934, p. 23. "The Regenerative Air Preheater," by C. E. Creutz, Combustion, March 1934, p. 33.

APPENDIX.

USE OF COMMON LOGARITHMS FOR SPECIAL CASES.

CASE I. TO DETERMINE THE nTH POWER OF A NUMBER LESS THAN UNITY.

Example: Find $0.5^{1.55}$ by logs.

In general $\log_{10} V^n = n \log_{10} V$; and in this case $V = 0.5$ and $n = 1.55$.

From the tables $\log_{10} 0.5 = 9.6990 - 10$,

Then, $1.55 \log_{10} 0.5 = 1.55\,(9.6990 - 10) = 15.033450 - 15.5$

Subtract 5.5 to reduce the negative part of the characteristic to 10, $\underline{5.5 \qquad - \;\; 5.5}$

Log. of answer $= 9.533450 - 10.0$

Corresponding number $= 0.342 = 0.5^{1.55}$.

(Note that a fraction raised to a power greater than unity gives a result less than the original fraction.)

CASE II. TO DETERMINE THE nTH ROOT OF A FRACTION.

Example: Given $V^{1.5} = 0.75$; Find V. Evidently, —

$$\log_{10} V = \log_{10}\left(\sqrt[1.5]{0.75}\right) = \log_{10}\left(0.75^{\frac{1}{1.5}}\right) = (\log_{10} 0.75) \div 1.5,$$

which is in the general form of $\log_{10} V = (\log_{10} C) \div n$,

where $C = 0.75$ and $n = 1.5$.

From the tables $\log_{10} 0.75 = 9.8751 - 10$.

Then $$\frac{(\log 0.75)}{1.5} = \frac{(9.8751 - 10)}{1.5} = 6.5833 - 6.666.$$

Add 3.334 to raise the negative part of the characteristic to 10, $\underline{3.3340 - \;\; 3.334}$

Log. of $V = 9.9173 - 10.000$

The corresponding number is 0.8266 which is $\sqrt[1.5]{0.75}$.

TABLE A. — COMMON LOGARITHMS (Log_{10}).

No.	0	1	2	3	4	5	6	7	8	9	Diff.
0	0	0000	3010	4771	6021	6990	7782	8451	9031	9542	
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	42
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	38
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	35
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	32
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	30
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	28
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	26
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	25
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	24
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	22
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	21
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	20
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	19
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	19
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	18
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	17
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	16
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	16
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	15
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	15
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	14
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	14
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	13
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	13
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	13
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	12
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	12
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	12
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	11
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	11
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	11
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	10
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	10
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	10
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	10
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	10
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	9
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	9
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	9
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	9
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	9
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	9
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	8
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	8
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	8

$e = 2.71828$

TABLE A. (*Concluded*). — COMMON LOGARITHMS (Log_{10}).

No.	0	1	2	3	4	5	6	7	8	9	Diff.
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	8
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	8
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	8
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	7
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	7
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	7
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	7
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	7
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	7
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	7
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	6
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	6
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	6
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	6
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	6
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	6
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	6
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	6
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	5
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	5
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	5
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	5
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	5
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	5
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	5
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	5
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	5
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	4

Napierian $\log_e = 2.302 \log_{10}$.

TABLE B.—NAPIERIAN LOGARITHMS ($\log_e$).
(Also Called Hyperbolic or Natural Logarithms)

No.	0	1	2	3	4	5	6	7	8	9	Avg. Diff.
1.0	0.0000	0100	0198	0296	0392	0488	0583	0677	0770	0862	95
1.1	0953	1044	1133	1222	1310	1398	1484	1570	1655	1740	87
1.2	1823	1906	1989	2070	2151	2231	2311	2390	2469	2546	80
1.3	2624	2700	2776	2852	2927	3001	3075	3148	3221	3293	74
1.4	3365	3436	3507	3577	3646	3716	3784	3853	3920	3988	69
1.5	0.4055	4121	4187	4253	4318	4383	4447	4511	4574	4637	65
1.6	4700	4762	4824	4886	4947	5008	5068	5128	5188	5247	61
1.7	5306	5365	5423	5481	5539	5596	5653	5710	5766	5822	57
1.8	5878	5933	5988	6043	6098	6152	6206	6259	6313	6366	54
1.9	6419	6471	6523	6575	6627	6678	6729	6780	6831	6881	51
2.0	0.6931	6981	7031	7080	7129	7178	7227	7275	7324	7372	49
2.1	7419	7467	7514	7561	7608	7655	7701	7747	7793	7839	47
2.2	7885	7930	7975	8020	8065	8109	8154	8198	8242	8286	44
2.3	8329	8372	8416	8459	8502	8544	8587	8629	8671	8713	43
2.4	8755	8796	8838	8879	8920	8961	9002	9042	9083	9123	41
2.5	0.9163	9203	9243	9282	9322	9361	9400	9439	9478	9517	39
2.6	9555	9594	9632	9670	9708	9746	9783	9821	9858	9895	38
2.7	9933	9969	1.0006	1.0043	1.0080	1.0116	1.0152	1.0188	1.0225	1.0260	36
2.8	1.0296	0332	0367	0403	0438	0473	0508	0543	0578	0613	35
2.9	0647	0682	0716	0750	0784	0818	0852	0886	0919	0953	34
3.0	1.0986	1019	1053	1086	1119	1151	1184	1217	1249	1282	33
3.1	1314	1346	1378	1410	1442	1474	1506	1537	1569	1600	32
3.2	1632	1663	1694	1725	1756	1787	1817	1848	1878	1909	31
3.3	1939	1969	2000	2030	2060	2090	2119	2149	2179	2208	30
3.4	2238	2267	2296	2326	2355	2384	2413	2442	2470	2499	29
3.5	1.2528	2556	2585	2613	2641	2669	2698	2726	2754	2782	28
3.6	2809	2837	2865	2892	2920	2947	2975	3002	3029	3056	27
3.7	3083	3110	3137	3164	3191	3218	3244	3271	3297	3324	27
3.8	3350	3376	3403	3429	3455	3481	3507	3533	3558	3584	26
3.9	3610	3635	3661	3686	3712	3737	3762	3788	3813	3838	25
4.0	1.3863	3888	3913	3938	3962	3987	4012	4036	4061	4085	25
4.1	4110	4134	4159	4183	4207	4231	4255	4279	4303	4327	24
4.2	4351	4375	4398	4422	4446	4469	4493	4516	4540	4563	23
4.3	4586	4609	4633	4656	4679	4702	4725	4748	4770	4793	23
4.4	4816	4839	4861	4884	4907	4929	4951	4974	4996	5019	22
4.5	1.5041	5063	5085	5107	5129	5151	5173	5195	5217	5239	22
4.6	5261	5282	5304	5326	5347	5369	5390	5412	5433	5454	21
4.7	5476	5497	5518	5539	5560	5581	5602	5623	5644	5665	21
4.8	5686	5707	5728	5748	5769	5790	5810	5831	5851	5872	20
4.9	5892	5913	5933	5953	5974	5994	6014	6034	6054	6074	20
5.0	1.6094	6114	6134	6154	6174	6194	6214	6233	6253	6273	20
5.1	6292	6312	6332	6351	6371	6390	6409	6429	6448	6467	19
5.2	6487	6506	6525	6544	6563	6582	6601	6620	6639	6658	19
5.3	6677	6696	6715	6734	6752	6771	6790	6808	6827	6845	18
5.4	6864	6882	6901	6919	6938	6956	6974	6993	7011	7029	18
5.5	1.7047	7066	7084	7102	7120	7138	7156	7174	7192	7210	18
5.6	7228	7246	7263	7281	7299	7317	7334	7352	7370	7387	18
5.7	7405	7422	7440	7457	7475	7492	7509	7527	7544	7561	17
5.8	7579	7596	7613	7630	7647	7664	7681	7699	7716	7733	17
5.9	7750	7766	7783	7800	7817	7834	7851	7867	7884	7901	17

TABLE B. (*Concluded*).—NAPIERIAN LOGARITHMS ($\log_e$).

No.	0	1	2	3	4	5	6	7	8	9	Avg. Diff.
6.0	1.7918	7934	7951	7967	7984	8001	8017	8034	8050	8066	16
6.1	8083	8099	8116	8132	8148	8165	8181	8197	8213	8229	16
6.2	8245	8262	8278	8294	8310	8326	8342	8358	8374	8390	16
6.3	8405	8421	8437	8453	8469	8485	8500	8516	8532	8547	16
6.4	8563	8579	8594	8610	8625	8641	8656	8672	8687	8703	15
6.5	1.8718	8733	8749	8764	8779	8795	8810	8825	8840	8856	15
6.6	8871	8886	8901	8916	8931	8946	8961	8976	8991	9006	15
6.7	9021	9036	9051	9066	9081	9095	9110	9125	9140	9155	15
6.8	9169	9184	9199	9213	9228	9242	9257	9272	9286	9301	15
6.9	9315	9330	9344	9359	9373	9387	9402	9416	9430	9445	14
7.0	1.9459	9473	9488	9502	9516	9530	9544	9559	9573	9587	14
7.1	9601	9615	9629	9643	9657	9671	9685	9699	9713	9727	14
7.2	9741	9755	9769	9782	9796	9810	9824	9838	9851	9865	14
7.3	9879	9892	9906	9920	9933	9947	9961	9974	9988	2.0001	13
7.4	2.0015	0028	0042	0055	0069	0082	0096	0109	0122	0136	13
7.5	2.0149	0162	0176	0189	0202	0215	0229	0242	0255	0268	13
7.6	0281	0295	0308	0321	0334	0347	0360	0373	0386	0399	13
7.7	0412	0425	0438	0451	0464	0477	0490	0503	0516	0528	13
7.8	0541	0554	0567	0580	0592	0605	0618	0631	0643	0656	13
7.9	0669	0681	0694	0707	0719	0732	0744	0757	0769	0782	12
8.0	2.0794	0807	0819	0832	0844	0857	0869	0882	0894	0906	12
8.1	0919	0931	0943	0956	0968	0980	0992	1005	1017	1029	12
8.2	1041	1054	1066	1078	1090	1102	1114	1126	1138	1150	12
8.3	1163	1175	1187	1199	1211	1223	1235	1247	1258	1270	12
8.4	1282	1294	1306	1318	1330	1342	1353	1365	1377	1389	12
8.5	2.1401	1412	1424	1436	1448	1459	1471	1483	1494	1506	12
8.6	1518	1529	1541	1552	1564	1576	1587	1599	1610	1622	12
8.7	1633	1645	1656	1668	1679	1691	1702	1713	1725	1736	11
8.8	1748	1759	1770	1782	1793	1804	1815	1827	1838	1849	11
8.9	1861	1872	1883	1894	1905	1917	1928	1939	1950	1961	11
9.0	2.1972	1983	1994	2006	2017	2028	2039	2050	2061	2072	11
9.1	2083	2094	2105	2116	2127	2138	2148	2159	2170	2181	11
9.2	2192	2203	2214	2225	2235	2246	2257	2268	2279	2289	11
9.3	2300	2311	2322	2332	2343	2354	2364	2375	2386	2396	11
9.4	2407	2418	2428	2439	2450	2460	2471	2481	2492	2502	11
9.5	2.2513	2523	2534	2544	2555	2565	2576	2586	2597	2607	10
9.6	2618	2628	2638	2649	2659	2670	2680	2690	2701	2711	10
9.7	2721	2732	2742	2752	2762	2773	2783	2793	2803	2814	10
9.8	2824	2834	2844	2854	2865	2875	2885	2895	2905	2915	10
9.9	2925	2935	2946	2956	2966	2976	2986	2996	3006	3016	10
10.0	2.3026	3125	3224	3322	3418	3513	3609	3702	3795	3887	9.5

$\log_{10} a = 0.4343 \log_e a$; $\log_e a = 2.3026 \log_{10} a$; $\log_e 0.1 = 0.6974 - 3$.

Examples: $\log_e 2150 = \log_e 2.15 + 3 \times \log_e 10 = 0.7655 + 3 \times 2.3026$

$\log_e 0.00215 = \log_e 2.15 + 3 \times \log_e 0.1 = 0.7655 + 3(0.6974 - 3)$

TABLE C_1.—PROPERTIES OF SATURATED STEAM AT LOW PRESSURES

Abridged from Keenan's Steam Tables, 1930, by permission of the publishers, The American Society of Mechanical Engineers.

For the saturation temperature and the heat content of the saturated liquid the values have been shortened to the nearest tenth of a unit.

Temp. Deg. Fahr.	Abs. Press. Inches Mercury	Abs. Press. Lb. per Sq. In.	Volume, Cu. Ft. per Lb.		Heat Content, B.t.u. per Lb.		Entropy, B.t.u. per Deg. Fahr. per Lb.		
Saturated Liquid and Vapor			Sat. Liquid	Sat. Vapor	Sat. Liquid	Sat. Vapor	Sat. Liquid	Vaporization	Sat. Vapor
$t_{sat.}$	p''	p	$\overline{V}_l$	$\overline{V}_{sat.}$	h_l	$h_{sat.}$	ϕ_l	ϕ_v	$\phi_{sat.}$
32.0	0.1806	0.0887	0.01602	3301.0	0.0	1073.4	0.0000	2.1834	2.1834
35.0	0.2036	0.1000	0.01602	2946.0	3.0	1074.8	0.0061	2.1668	2.1729
40.0	0.2478	0.1217	0.01602	2445.0	8.1	1077.1	0.0162	2.1397	2.1559
45.0	0.3003	0.1475	0.01602	2037.2	13.1	1079.4	0.0262	2.1131	2.1393
50.0	0.3624	0.1780	0.01602	1704.5	18.1	1081.7	0.0361	2.0870	2.1231
55.0	0.4357	0.2140	0.01603	1431.8	23.1	1084.0	0.0459	2.0615	2.1073
58.8	0.5000	0.2456	0.01603	1256.9	26.9	1085.7	0.0533	2.0422	2.0955
60.0	0.5214	0.2561	0.01603	1208.0	28.1	1086.2	0.0555	2.0364	2.0919
62.0	0.5597	0.2749	0.01603	1129.4	30.1	1087.1	0.0594	2.0265	2.0859
64.0	0.6004	0.2949	0.01604	1057.0	32.0	1088.0	0.0632	2.0167	2.0799
66.0	0.6437	0.3162	0.01604	989.6	34.0	1089.0	0.0670	2.0069	2.0739
68.0	0.6897	0.3388	0.01605	927.1	36.0	1089.8	0.0708	1.9973	2.0680
70.0	0.7386	0.3628	0.01605	869.0	38.0	1090.8	0.0746	1.9877	2.0622
70.4	0.7500	0.3684	0.01605	856.5	38.5	1091.0	0.0754	1.9856	2.0609
72.0	0.7905	0.3883	0.01606	815.0	40.0	1091.7	0.0783	1.9782	2.0565
74.0	0.8455	0.4153	0.01606	764.8	42.0	1092.6	0.0820	1.9687	2.0507
76.0	0.9039	0.4440	0.01606	718.0	44.0	1093.5	0.0858	1.9593	2.0451
78.0	0.9658	0.4744	0.01607	674.5	46.0	1094.4	0.0895	1.9500	2.0395
79.1	1.0000	0.4912	0.01607	652.7	47.1	1094.9	0.0914	1.9451	2.0365
80.0	1.0316	0.5067	0.01607	633.8	48.0	1095.3	0.0932	1.9407	2.0340
82.0	1.1012	0.5409	0.01608	595.9	50.0	1096.2	0.0969	1.9316	2.0285
84.0	1.1751	0.5772	0.01608	560.5	52.0	1097.1	0.1006	1.9224	2.0230
86.0	1.2527	0.6153	0.01609	527.7	54.0	1098.0	0.1042	1.9134	2.0176
88.0	1.3345	0.6555	0.01609	497.1	56.0	1098.9	0.1079	1.9044	2.0123
90.0	1.4210	0.6980	0.01610	468.5	58.0	1099.8	0.1115	1.8955	2.0070
91.7	1.5000	0.7368	0.01610	445.3	59.7	1100.6	0.1147	1.8877	2.0024
95.0	1.6590	0.8149	0.01611	404.9	63.0	1102.0	0.1206	1.8734	1.9940
100.0	1.9314	0.9487	0.01613	350.8	67.9	1104.2	0.1295	1.8518	1.9813
101.2	2.0000	0.9824	0.01613	339.5	69.1	1104.8	0.1316	1.8468	1.9784
108.7	2.5000	1.2280	0.01616	275.2	76.6	1108.1	0.1450	1.8148	1.9598
115.1	3.0000	1.4736	0.01618	231.8	83.0	1110.8	0.1561	1.7885	1.9446

TABLE C.—PROPERTIES OF SATURATED STEAM

Abridged from Keenan's Steam Tables, 1930, by permission of the publishers, The American Society of Mechanical Engineers.

For the saturation temperature and the heat content of the saturated liquid the values have been shortened to the nearest tenth of a unit.

Abs. Press. Lb. per Sq. In.	Temp. Deg. Fahr.	Volume, Cu. Ft. per Lb.		Heat Content, B.t.u. per Lb.		Total Latent Heat of Vaporization, B.t.u. per Lb.	Entropy, B.t.u. per Deg. Fahr. per Lb.		
Saturated Liquid and Vapor		Sat. Liquid	Sat. Vapor	Sat. Liquid	Sat. Vapor		Sat. Liquid	Vaporization	Sat. Vapor
p	$t_{\text{sat.}}$	$\overline{V}_l$	$\overline{V}_{\text{sat.}}$	h_l	$h_{\text{sat.}}$	r	ϕ_l	ϕ_v	$\phi_{\text{sat.}}$
0.0887	32	0.01602	3301	0.0	1073.4	1073.4	0.0000	2.1834	2.1834
0.1217	40	0.01602	2445	8.1	1077.1	1069.1	0.0162	2.1397	2.1559
0.1780	50	0.01602	1704.8	18.1	1081.7	1063.6	0.0361	2.0870	2.1231
0.2456 (½″ Hg)	58.8	0.01603	1256.9	26.9	1085.7	1058.8	0.0533	2.0422	2.0955
0.2561	60	0.01603	1208.0	28.0	1086.2	1058.2	0.0555	2.0364	2.0919
0.3628	70	0.01605	869.0	38.0	1090.8	1052.7	0.0746	1.9877	2.0622
0.4912 (1″ Hg)	79.1	0.01607	652.7	47.1	1094.9	1047.8	0.0914	1.9451	2.0365
0.5067	80	0.01607	633.8	48.0	1095.3	1047.3	0.0932	1.9407	2.0340
0.6980	90	0.01610	468.5	58.0	1099.8	1041.8	0.1115	1.8955	2.0070
0.7368 (1½″Hg)	91.7	0.01610	445.3	59.7	1100.6	1040.8	0.1147	1.8877	2.0024
0.9487	100	0.01613	350.8	67.9	1104.2	1036.3	0.1295	1.8518	1.9813
0.9824 (2″ Hg)	101.2	0.01613	339.5	69.1	1104.8	1035.7	0.1316	1.8468	1.9784
1	101.8	0.01614	333.9	69.7	1105.0	1035.3	0.1326	1.8442	1.9769
1.274	110	0.01616	265.8	77.9	1108.6	1030.8	0.1472	1.8095	1.9567
1.470	115	0.01618	232.3	82.9	1110.8	1027.9	0.1559	1.7889	1.9448
1.692	120	0.01620	203.6	87.9	1113.0	1025.1	0.1646	1.7685	1.9331
2	126.1	0.01623	173.96	94.0	1115.6	1021.6	0.1750	1.7442	1.9192
3	141.5	0.01630	118.86	109.3	1122.0	1012.7	0.2009	1.6847	1.8856
4	153.0	0.01636	90.74	120.8	1126.8	1005.9	0.2198	1.6420	1.8618
5	162.2	0.01641	73.61	130.1	1130.6	1000.4	0.2348	1.6088	1.8435
6	170.1	0.01645	62.05	137.9	1133.7	995.8	0.2473	1.5814	1.8287
7	176.8	0.01649	53.70	144.7	1136.4	991.7	0.2580	1.5582	1.8162
8	182.9	0.01652	47.39	150.8	1138.9	988.1	0.2674	1.5379	1.8053
9	188.3	0.01656	42.44	156.2	1141.0	984.8	0.2758	1.5200	1.7958
10	193.2	0.01658	38.45	161.1	1143.0	981.8	0.2834	1.5040	1.7874
11	197.7	0.01661	35.17	165.7	1144.8	979.1	0.2903	1.4894	1.7797
12	202.0	0.01664	32.42	169.9	1146.4	976.5	0.2968	1.4760	1.7727
13	205.9	0.01666	30.08	173.8	1147.9	974.1	0.3027	1.4636	1.7663
14	209.6	0.01669	28.06	177.5	1149.3	971.8	0.3082	1.4521	1.7604
14.696	212.0	0.01670	26.82	180.0	1150.2	970.2	0.3119	1.4446	1.7564

TABLE C.—PROPERTIES OF SATURATED STEAM.—(*Continued*)

Abs. Press.	Temp.	Volume, Cu. Ft. per Lb.		Heat Content, B.t.u. per Lb.		Total Latent Heat of Vaporization, B.t.u. per Lb.	Entropy, B.t.u. per Deg. Fahr. per Lb.		
Lb. per Sq. In.	Deg. Fahr.								
Saturated Liquid and Vapor		Sat. Liquid	Sat. Vapor	Sat. Liquid	Sat. Vapor		Sat. Liquid	Vaporization	Sat. Vapor
p	$t_{sat.}$	$\overline{V}_l$	$\overline{V}_{sat.}$	h_l	$h_{sat.}$	r	ϕ_l	ϕ_v	$\phi_{sat.}$
15	213.0	0.01671	26.31	181.0	1150.6	969.6	0.3134	1.4414	1.7548
16	216.3	0.01673	24.76	184.4	1151.8	967.4	0.3184	1.4312	1.7496
17	219.4	0.01676	23.40	187.5	1152.9	965.4	0.3230	1.4218	1.7448
18	222.4	0.01678	22.18	190.5	1154.0	963.5	0.3274	1.4127	1.7402
19	225.2	0.01680	21.08	193.3	1155.0	961.7	0.3316	1.4042	1.7358
20	228.0	0.01682	20.095	196.1	1156.0	959.9	0.3356	1.3960	1.7317
21	230.6	0.01684	19.197	198.7	1156.9	958.2	0.3395	1.3883	1.7278
22	233.1	0.01685	18.380	201.2	1157.8	956.6	0.3431	1.3809	1.7240
23	235.5	0.01687	17.630	203.7	1158.6	955.0	0.3466	1.3738	1.7204
24	237.8	0.01689	16.941	206.1	1159.5	953.4	0.3500	1.3670	1.7170
25	240.1	0.01690	16.306	208.3	1160.2	951.9	0.3533	1.3604	1.7137
26	242.2	0.01692	15.718	210.5	1161.0	950.4	0.3564	1.3542	1.7106
27	244.4	0.01694	15.172	212.7	1161.7	949.0	0.3594	1.3481	1.7075
28	246.4	0.01695	14.664	214.7	1162.4	947.7	0.3624	1.3422	1.7046
29	248.4	0.01697	14.190	216.8	1163.1	946.3	0.3652	1.3365	1.7018
30	250.3	0.01698	13.745	218.7	1163.7	945.0	0.3680	1.3310	1.6990
32	254.0	0.01701	12.940	222.5	1165.0	942.5	0.3732	1.3206	1.6938
34	257.6	0.01704	12.226	226.1	1166.1	940.0	0.3783	1.3107	1.6890
36	260.9	0.01707	11.587	229.5	1167.2	937.7	0.3830	1.3014	1.6844
38	264.2	0.01710	11.015	232.8	1168.3	935.5	0.3876	1.2925	1.6800
40	267.2	0.01712	10.497	235.9	1169.2	933.3	0.3919	1.2840	1.6759
42	270.2	0.01715	10.027	239.0	1170.2	931.2	0.3961	1.2759	1.6720
44	273.1	0.01717	9.599	241.9	1171.1	929.2	0.4000	1.2682	1.6683
46	275.8	0.01719	9.207	244.7	1171.9	927.2	0.4039	1.2608	1.6647
48	278.4	0.01722	8.846	247.4	1172.7	925.4	0.4076	1.2537	1.6613
50	281.0	0.01724	8.514	250.0	1173.5	923.5	0.4111	1.2469	1.6580
52	283.5	0.01726	8.206	252.5	1174.3	921.7	0.4145	1.2404	1.6549
54	285.9	0.01728	7.919	255.0	1175.0	920.0	0.4178	1.2340	1.6518
56	288.2	0.01730	7.653	257.4	1175.7	918.3	0.4210	1.2279	1.6489
58	290.5	0.01732	7.405	259.7	1176.4	916.6	0.4241	1.2220	1.6461
60	292.7	0.01735	7.172	262.0	1177.0	915.0	0.4271	1.2162	1.6434
62	294.8	0.01737	6.955	264.2	1177.6	913.4	0.4300	1.2107	1.6407
64	296.9	0.01739	6.749	266.3	1178.2	911.9	0.4329	1.2053	1.6382
66	299.0	0.01741	6.556	268.4	1178.8	910.4	0.4356	1.2001	1.6357
68	301.0	0.01743	6.375	270.5	1179.4	908.9	0.4384	1.1950	1.6333
70	302.9	0.01744	6.203	272.5	1179.9	907.4	0.4410	1.1900	1.6310
72	304.8	0.01746	6.041	274.5	1180.5	906.0	0.4435	1.1852	1.6287
74	306.7	0.01748	5.887	276.4	1181.0	904.6	0.4460	1.1805	1.6265
76	308.5	0.01750	5.741	278.3	1181.5	903.2	0.4485	1.1759	1.6244
78	310.3	0.01752	5.602	280.1	1182.0	901.9	0.4509	1.1714	1.6223

TABLE C.—PROPERTIES OF SATURATED STEAM.—(*Continued*)

Abs. Press. Lb. per Sq. In.	Temp. Deg. Fahr.	Volume, Cu. Ft. per Lb.		Heat Content, B.t.u. per Lb.		Total Latent Heat of Vaporization, B.t.u. per Lb.	Entropy, B.t.u. per Deg. Fahr. per Lb.		
Saturated Liquid and Vapor		Sat. Liquid	Sat. Vapor	Sat. Liquid	Sat. Vapor		Sat. Liquid	Vaporization	Sat. Vapor
p	$t_{sat.}$	$\overline{V}_l$	$\overline{V}_{sat.}$	h_l	$h_{sat.}$	r	ϕ_l	ϕ_v	$\phi_{sat.}$
80	312.0	0.01754	5.470	281.9	1182.4	900.5	0.4532	1.1670	1.6202
82	313.7	0.01756	5.343	283.7	1182.9	899.2	0.4555	1.1627	1.6182
84	315.4	0.01757	5.222	285.4	1183.4	897.9	0.4578	1.1586	1.6163
86	317.1	0.01759	5.107	287.1	1183.8	896.7	0.4599	1.1545	1.6144
88	318.7	0.01761	4.997	288.8	1184.2	895.4	0.4621	1.1505	1.6126
90	320.3	0.01763	4.892	290.4	1184.6	894.2	0.4642	1.1465	1.6107
92	321.8	0.01764	4.791	292.1	1185.0	893.0	0.4663	1.1427	1.6090
94	323.4	0.01766	4.694	293.7	1185.4	891.8	0.4683	1.1389	1.6072
96	324.9	0.01768	4.602	295.2	1185.8	890.6	0.4703	1.1352	1.6055
98	326.4	0.01769	4.512	296.8	1186.2	889.4	0.4723	1.1316	1.6038
100	327.8	0.01771	4.426	298.3	1186.6	888.2	0.4742	1.1280	1.6022
102	329.3	0.01773	4.344	299.8	1186.9	887.1	0.4761	1.1245	1.6006
104	330.7	0.01774	4.265	301.3	1187.3	886.0	0.4779	1.1211	1.5990
106	332.1	0.01776	4.189	302.8	1187.6	884.9	0.4798	1.1177	1.5974
108	333.4	0.01777	4.115	304.2	1188.0	883.8	0.4816	1.1144	1.5959
110	334.8	0.01779	4.044	305.6	1188.3	882.7	0.4834	1.1111	1.5944
115	338.1	0.01783	3.878	309.0	1189.1	880.0	0.4876	1.1032	1.5908
120	341.3	0.01786	3.725	312.4	1189.8	877.4	0.4918	1.0956	1.5874
125	344.3	0.01790	3.587	315.6	1190.5	874.9	0.4958	1.0882	1.5840
130	347.3	0.01794	3.451	318.7	1191.2	872.4	0.4996	1.0812	1.5808
135	350.2	0.01797	3.329	321.8	1191.8	870.0	0.5034	1.0743	1.5777
140	353.0	0.01801	3.216	324.7	1192.4	867.7	0.5070	1.0677	1.5747
145	355.8	0.01804	3.110	327.6	1193.0	865.3	0.5105	1.0612	1.5718
150	358.4	0.01808	3.010	330.4	1193.5	863.1	0.5140	1.0550	1.5690
155	361.0	0.01811	2.917	333.2	1194.0	860.8	0.5173	1.0490	1.5663
160	363.5	0.01814	2.830	335.9	1194.5	858.7	0.5205	1.0431	1.5636
165	366.0	0.01818	2.748	338.5	1195.0	856.5	0.5237	1.0374	1.5611
170	368.4	0.01821	2.671	341.0	1195.4	854.4	0.5268	1.0318	1.5586
175	370.8	0.01824	2.598	343.5	1195.9	852.3	0.5298	1.0264	1.5562
180	373.1	0.01827	2.529	346.0	1196.3	850.3	0.5327	1.0211	1.5538
185	375.3	0.01830	2.463	348.4	1196.7	848.2	0.5356	1.0159	1.5515
190	377.5	0.01833	2.401	350.8	1197.0	846.3	0.5384	1.0109	1.5493
195	379.7	0.01836	2.342	353.1	1197.4	844.3	0.5411	1.0060	1.5471
200	381.8	0.01839	2.285	355.3	1197.8	842.4	0.5438	1.0012	1.5450
210	385.9	0.01844	2.180	359.8	1198.4	838.6	0.5491	0.9918	1.5409
220	389.9	0.01850	2.084	364.0	1199.0	835.0	0.5540	0.9829	1.5369
230	393.7	0.01856	1.9964	368.1	1199.6	831.4	0.5588	0.9743	1.5332
240	397.4	0.01861	1.9156	372.1	1200.1	827.9	0.5635	0.9661	1.5295
250	401.0	0.01867	1.8410	376.0	1200.5	824.5	0.5680	0.9581	1.5261
260	404.4	0.01872	1.7723	379.8	1201.0	821.2	0.5723	0.9504	1.5227

TABLE C.—PROPERTIES OF SATURATED STEAM.—(*Continued*)

Abs. Press. Lb. per Sq. In.	Temp. Deg. Fahr.	Volume, Cu. Ft. per Lb.		Heat Content, B.t.u. per Lb.		Total Latent Heat of Vaporization, B.t.u. per Lb.	Entropy, B.t.u. per Deg. Fahr. per Lb.		
Saturated Liquid and Vapor		Sat. Liquid	Sat. Vapor	Sat. Liquid	Sat. Vapor		Sat. Liquid	Vaporization	Sat. Vapor
p	$t_{sat.}$	$\overline{V}_l$	$\overline{V}_{sat.}$	h_l	$h_{sat.}$	r	ϕ_l	ϕ_v	$\phi_{sat.}$
270	407.8	0.01877	1.7083	383.4	1201.4	818.0	0.5765	0.9430	1.5194
280	411.1	0.01882	1.6490	387.0	1201.8	814.7	0.5805	0.9357	1.5163
290	414.2	0.01887	1.5934	390.5	1202.1	811.6	0.5845	0.9287	1.5132
300	417.3	0.01892	1.5414	393.9	1202.4	808.5	0.5883	0.9220	1.5102
320	423.3	0.01901	1.4469	400.5	1203.0	802.5	0.5957	0.9089	1.5046
340	429.0	0.01910	1.3630	406.8	1203.4	796.6	0.6027	0.8965	1.4992
360	434.4	0.01918	1.2881	412.8	1203.7	790.9	0.6094	0.8846	1.4940
380	439.6	0.01927	1.2208	418.6	1203.9	785.3	0.6157	0.8733	1.4891
400	444.6	0.0194	1.1601	424.2	1204.1	779.8	0.6218	0.8625	1.4843
420	449.4	0.0194	1.1047	429.6	1204.1	774.5	0.6277	0.8520	1.4798
440	454.0	0.0195	1.0540	434.8	1204.1	769.3	0.6334	0.8420	1.4753
460	458.5	0.0196	1.0077	439.9	1204.0	764.1	0.6388	0.8322	1.4711
480	462.8	0.0197	0.9653	444.9	1203.9	759.0	0.6441	0.8228	1.4670
500	467.0	0.0198	0.9261	449.7	1203.7	754.0	0.6493	0.8137	1.4630
600	486.2	0.0202	0.7677	472.3	1202.1	729.8	0.6731	0.7716	1.4447
700	503.0	0.0206	0.6527	492.9	1199.7	706.8	0.6943	0.7342	1.4285
800	518.2	0.0209	0.5653	511.8	1196.7	684.9	0.7135	0.7004	1.4139
900	531.9	0.0213	0.4969	529.5	1193.3	663.8	0.7311	0.6694	1.4005
1000	544.6	0.0217	0.4419	546.0	1189.6	643.5	0.7473	0.6408	1.3881
1100	556.3	0.0222	0.3960	561.7	1185.6	623.9	0.7624	0.6141	1.3765
1200	567.1	0.0226	0.3582	576.5	1181.4	604.9	0.7764	0.5891	1.3656
1300	577.3	0.0230	0.3259	590.6	1177.0	586.3	0.7897	0.5654	1.3552
1400	587.0	0.0235	0.2983	604.3	1172.4	568.1	0.8024	0.5428	1.3452
1500	596.1	0.0239	0.2741	617.5	1167.6	550.2	0.8146	0.5212	1.3357
1600	604.7	0.0244	0.2528	630.2	1162.7	532.6	0.8262	0.5003	1.3265
1800	620.9	0.0254	0.2167	654.7	1151.8	497.2	0.8482	0.4601	1.3083
2000	635.6	0.0265	0.1875	679.0	1139.0	460.0	0.8696	0.4200	1.2896
2200	649.2	0.0277	0.1623	703.7	1123.8	420.0	0.8912	0.3788	1.2700
2400	661.9	0.0292	0.1404	729.4	1105.8	376.4	0.9133	0.3356	1.2488
2600	673.8	0.0310	0.1205	756.7	1084.5	327.8	0.9364	0.2892	1.2257
2800	684.9	0.0333	0.1021	786.7	1058.9	272.3	0.9618	0.2379	1.1996
3000	695.2	0.0367	0.0844	823.1	1025.6	202.5	0.9922	0.1754	1.1676
3200	704.9	0.0459	0.0601	887.0	962.9	75.9	1.0461	0.0651	1.1112
3226*	706.1	0.0522	0.0522	925.0	925.0	0.0	1.0785	0.00	1.0785

* The last line gives the critical values.

TABLE D.—SUPERHEATED STEAM

Abridged from Keenan's Steam Tables, 1930, by permission of the publishers, The American Society of Mechanical Engineers.

Abs. Press., Lb. per Sq. In. (Sat. Temp.)	Symbol	Temperature, Degrees Fahrenheit							
		150	200	300	400	500	600	800	1000
1 (101.76)	$\overline{V}$	362.7	392.5	452.1	511.7	571.3	630.9	750.2	869.4
	h	1127.0	1149.8	1195.0	1240.7	1287.2	1334.6	1432.6	1535.2
	ϕ	2.0144	2.0503	2.1142	2.1707	2.2218	2.2688	2.3535	2.4291
2 (126.10)	$\overline{V}$	181.2	196.08	225.9	255.7	285.6	315.4	375.1	434.7
	h	1126.5	1149.4	1194.8	1240.6	1287.1	1334.6	1432.5	1535.2
	ϕ	1.9374	1.9734	2.0376	2.0942	2.1454	2.1924	2.2771	2.3527
5 (162.25)	$\overline{V}$	....	78.17	90.21	102.19	114.16	126.11	149.99	173.83
	h	....	1148.2	1194.2	1240.2	1286.9	1334.4	1432.5	1535.1
	ϕ	....	1.8712	1.9361	1.9930	2.0443	2.0914	2.1761	2.2517
10 (193.21)	$\overline{V}$	....	38.88	44.98	51.01	57.02	63.01	74.96	86.89
	h	....	1146.3	1193.2	1239.6	1286.4	1334.0	1432.3	1535.0
	ϕ	....	1.7925	1.8587	1.9161	1.9676	2.0148	2.0997	2.1753
15 (213.03)	$\overline{V}$	....	....	29.90	33.95	37.97	41.98	49.95	57.91
	h	....	....	1192.2	1238.9	1286.0	1333.7	1432.1	1534.9
	ϕ	....	....	1.8130	1.8708	1.9226	1.9699	2.0550	2.1307
20 (227.96)	$\overline{V}$	....	....	22.36	25.43	28.45	31.46	37.44	43.42
	h	....	....	1191.1	1238.3	1285.5	1333.4	1432.0	1534.8
	ϕ	....	....	1.7802	1.8386	1.8906	1.9380	2.0232	2.0989
25 (240.07)	$\overline{V}$	....	....	17.838	20.30	22.73	25.15	29.94	34.73
	h	....	....	1190.0	1237.6	1285.1	1333.0	1431.8	1534.7
	ϕ	....	....	1.7544	1.8134	1.8657	1.9132	1.9985	2.0743
30 (250.34)	$\overline{V}$	....	....	14.818	16.890	18.923	20.94	24.94	28.93
	h	....	....	1188.7	1237.0	1284.6	1332.7	1431.6	1534.6
	ϕ	....	....	1.7332	1.7928	1.8452	1.8928	1.9783	2.0541
40 (267.24)	$\overline{V}$	....	....	11.044	12.623	14.161	15.682	18.686	21.68
	h	....	....	1186.5	1235.6	1283.7	1332.0	1431.3	1534.4
	ϕ	....	....	1.6990	1.7599	1.8128	1.8607	1.9464	2.0224
50 (281.01)	$\overline{V}$	....	....	8.777	10.061	11.304	12.527	14.934	17.337
	h	....	....	1183.9	1234.2	1282.8	1331.4	1431.0	1534.1
	ϕ	....	....	1.6718	1.7341	1.7875	1.8357	1.9217	1.9977
60 (292.71)	$\overline{V}$	....	....	7.260	8.353	9.398	10.423	12.436	14.440
	h	....	....	1181.2	1232.8	1281.9	1330.7	1430.6	1533.9
	ϕ	....	....	1.6488	1.7128	1.7667	1.8151	1.9014	1.9775
70 (302.92)	$\overline{V}$	....	....	....	7.132	8.036	8.920	10.651	12.370
	h	....	....	....	1231.4	1280.9	1330.0	1430.3	1533.7
	ϕ	....	....	....	1.6945	1.7490	1.7976	1.8843	1.9604
80 (312.03)	$\overline{V}$	....	....	....	6.217	7.015	7.793	9.313	10.817
	h	....	....	....	1229.9	1280.0	1329.3	1429.9	1533.5
	ϕ	....	....	....	1.6785	1.7336	1.7824	1.8694	1.9456

TABLE D.—SUPERHEATED STEAM.—(*Continued*)

Abs. Press., Lb. per Sq. In. (Sat. Temp.)	Symbol	Temperature, Degrees Fahrenheit							
		400	450	500	600	700	800	900	1000
100	$\bar{V}$	4.934	5.263	5.585	6.215	6.831	7.439	8.044	8.644
(327.83)	h	1226.9	1252.8	1278.0	1327.9	1378.2	1429.2	1480.7	1533.1
	ϕ	1.6512	1.6805	1.7075	1.7569	1.8023	1.8445	1.8838	1.9209
120	$\bar{V}$	4.077	4.359	4.632	5.162	5.680	6.189	6.693	7.196
(341.26)	h	1223.8	1250.4	1276.1	1326.5	1377.2	1428.5	1480.2	1532.7
	ϕ	1.6283	1.6584	1.6859	1.7359	1.7816	1.8240	1.8635	1.9007
140	$\bar{V}$	3.465	3.713	3.951	4.410	4.857	5.297	5.728	6.161
(353.03)	h	1220.5	1247.9	1274.1	1325.1	1376.2	1427.8	1479.7	1532.2
	ϕ	1.6084	1.6393	1.6674	1.7179	1.7640	1.8066	1.8462	1.8836
160	$\bar{V}$	3.005	3.227	3.440	3.846	4.240	4.627	5.006	5.385
(363.55)	h	1217.1	1245.4	1272.1	1323.7	1375.2	1427.1	1479.1	1531.8
	ϕ	1.5906	1.6224	1.6510	1.7022	1.7487	1.7915	1.8313	1.8687
180	$\bar{V}$	2.646	2.849	3.041	3.407	3.760	4.105	4.444	4.782
(373.08)	h	1213.5	1242.7	1270.0	1322.2	1374.2	1426.3	1478.5	1531.4
	ϕ	1.5742	1.6073	1.6364	1.6882	1.7351	1.7782	1.8180	1.8555
200	$\bar{V}$	2.358	2.547	2.722	3.056	3.376	3.688	3.995	4.299
(381.82)	h	1209.8	1240.0	1267.9	1320.8	1373.1	1425.6	1478.0	1531.0
	ϕ	1.5592	1.5934	1.6231	1.6756	1.7228	1.7662	1.8062	1.8438
220	$\bar{V}$	2.122	2.299	2.462	2.769	3.062	3.347	3.628	3.903
(389.89)	h	1205.9	1237.3	1265.7	1319.3	1372.1	1424.8	1477.4	1530.6
	ϕ	1.5450	1.5805	1.6109	1.6641	1.7117	1.7553	1.7954	1.8331
250	$\bar{V}$	....	2.001	2.149	2.424	2.685	2.938	3.186	3.429
(400.97)	h	....	1232.9	1262.4	1317.1	1370.5	1423.7	1476.6	1530.0
	ϕ	....	1.5627	1.5944	1.6483	1.6966	1.7406	1.7810	1.8188
300	$\bar{V}$	....	1.6347	1.7648	2.002	2.224	2.438	2.646	2.849
(417.33)	h	....	1225.3	1256.7	1313.3	1367.8	1421.7	1475.1	1528.9
	ϕ	....	1.5359	1.5695	1.6256	1.6747	1.7193	1.7601	1.7983
350	$\bar{V}$	....	1.3712	1.4899	1.7003	1.8945	2.080	2.260	2.435
(431.71)	h	....	1217.1	1250.7	1309.4	1365.1	1419.8	1473.6	1527.9
	ϕ	....	1.5117	1.5477	1.6059	1.6561	1.7012	1.7424	1.7809
400	$\bar{V}$	....	1.1726	1.2828	1.4740	1.6472	1.8119	1.9704	2.125
(444.58)	h	....	1208.3	1244.3	1305.5	1362.3	1417.7	1472.1	1526.8
	ϕ	....	1.4892	1.5276	1.5884	1.6396	1.6854	1.7270	1.7658
450	$\bar{V}$	....	....	1.1204	1.2972	1.4548	1.6032	1.7455	1.8834
(456.27)	h	....	....	1237.6	1301.5	1359.4	1415.7	1470.6	1525.8
	ϕ	....	....	1.5091	1.5725	1.6248	1.6714	1.7133	1.7524
500	$\bar{V}$	....	....	0.9905	1.1558	1.3009	1.4365	1.5655	1.6903
(466.99)	h	....	....	1230.5	1297.3	1356.6	1413.6	1469.1	1524.8
	ϕ	....	....	1.4915	1.5579	1.6113	1.6586	1.7009	1.7404

TABLE D.—SUPERHEATED STEAM.—(*Continued*)

Abs. Press., Lb. per Sq. In. (Sat. Temp.)	Symbol	Temperature, Degrees Fahrenheit							
		550	600	650	700	750	800	900	1000
550 (476.92)	$\overline{V}$	0.9658	1.0398	1.1089	1.1746	1.2381	1.2996	1.4181	1.5321
	h	1260.3	1293.1	1323.8	1353.6	1382.8	1411.5	1467.6	1523.8
	ϕ	1.5126	1.5443	1.5726	1.5990	1.6236	1.6468	1.6897	1.7296
600 (486.17)	$\overline{V}$	0.8728	0.9431	1.0080	1.0694	1.1285	1.1855	1.2953	1.4003
	h	1254.6	1288.7	1320.2	1350.6	1380.3	1409.3	1466.1	1522.8
	ϕ	1.4986	1.5316	1.5607	1.5874	1.6125	1.6360	1.6794	1.7196
650 (494.86)	$\overline{V}$	0.7933	0.8609	0.9224	0.9802	1.0355	1.0890	1.1917	1.2886
	h	1248.6	1284.3	1316.6	1347.6	1376.8	1407.1	1464.6	1521.8
	ϕ	1.4851	1.5195	1.5494	1.5766	1.6021	1.6259	1.6698	1.7103
700 (503.04)	$\overline{V}$	0.7251	0.7905	0.8490	0.9038	0.9559	1.0063	1.1029	1.1929
	h	1242.4	1279.7	1312.9	1344.5	1375.1	1404.9	1463.0	1520.8
	ϕ	1.4720	1.5080	1.5387	1.5665	1.5923	1.6165	1.6608	1.7018
750 (510.80)	$\overline{V}$	0.6652	0.7289	0.7852	0.8373	0.8869	0.9348	1.0254	1.1100
	h	1235.8	1275.0	1309.2	1341.4	1372.4	1402.7	1461.5	1519.8
	ϕ	1.4591	1.4970	1.5285	1.5569	1.5831	1.6076	1.6524	1.6938
800 (518.18)	$\overline{V}$	0.6128	0.6750	0.7293	0.7791	0.8265	0.8723	0.9577	1.0374
	h	1228.8	1270.1	1305.4	1338.2	1369.7	1400.4	1459.9	1518.8
	ϕ	1.4462	1.4862	1.5187	1.5477	1.5744	1.5992	1.6446	1.6864
900 (531.95)	$\overline{V}$	0.5234	0.5844	0.6356	0.6821	0.7258	0.7675	0.8451	0.9166
	h	1213.6	1259.8	1297.4	1331.7	1364.2	1395.8	1456.8	1516.8
	ϕ	1.4208	1.4656	1.5003	1.5304	1.5579	1.5835	1.6301	1.6727
1000 (544.58)	$\overline{V}$	0.4495	0.5111	0.5602	0.6040	0.6449	0.6837	0.7547	0.8199
	h	1196.5	1248.7	1289.2	1324.9	1358.5	1391.0	1453.6	1514.8
	ϕ	1.3949	1.4455	1.4829	1.5144	1.5427	1.5691	1.6169	1.6603
1100 (556.28)	$\overline{V}$	....	0.4500	0.4983	0.5401	0.5788	0.6152	0.6810	0.7408
	h	....	1236.6	1280.5	1317.9	1352.6	1386.1	1450.4	1512.8
	ϕ	....	1.4257	1.4662	1.4993	1.5285	1.5557	1.6048	1.6491
1200 (567.14)	$\overline{V}$	....	0.3985	0.4462	0.4865	0.5233	0.5578	0.6195	0.6750
	h	....	1223.4	1271.3	1310.6	1346.5	1381.1	1447.2	1510.8
	ϕ	....	1.4058	1.4501	1.4848	1.5150	1.5431	1.5937	1.6388
1300 (577.32)	$\overline{V}$	....	0.3537	0.4018	0.4412	0.4767	0.5095	0.5675	0.6190
	h	....	1208.7	1261.6	1303.0	1340.2	1376.0	1444.0	1508.8
	ϕ	....	1.3855	1.4343	1.4709	1.5022	1.5312	1.5832	1.6292
1400 (586.96)	$\overline{V}$	....	0.3144	0.3634	0.4021	0.4365	0.4678	0.5229	0.5712
	h	....	1192.4	1251.2	1295.2	1333.8	1370.8	1440.8	1506.9
	ϕ	....	1.3643	1.4187	1.4574	1.4899	1.5199	1.5736	1.6204
1500 (596.08)	$\overline{V}$	....	0.2789	0.3298	0.3682	0.4014	0.4318	0.4840	0.5298
	h	....	1174.3	1240.2	1287.1	1327.2	1365.4	1437.6	1504.9
	ϕ	....	1.3420	1.4029	1.4442	1.4780	1.5091	1.5642	1.6120

TABLE D.—SUPERHEATED STEAM.—(*Continued*)

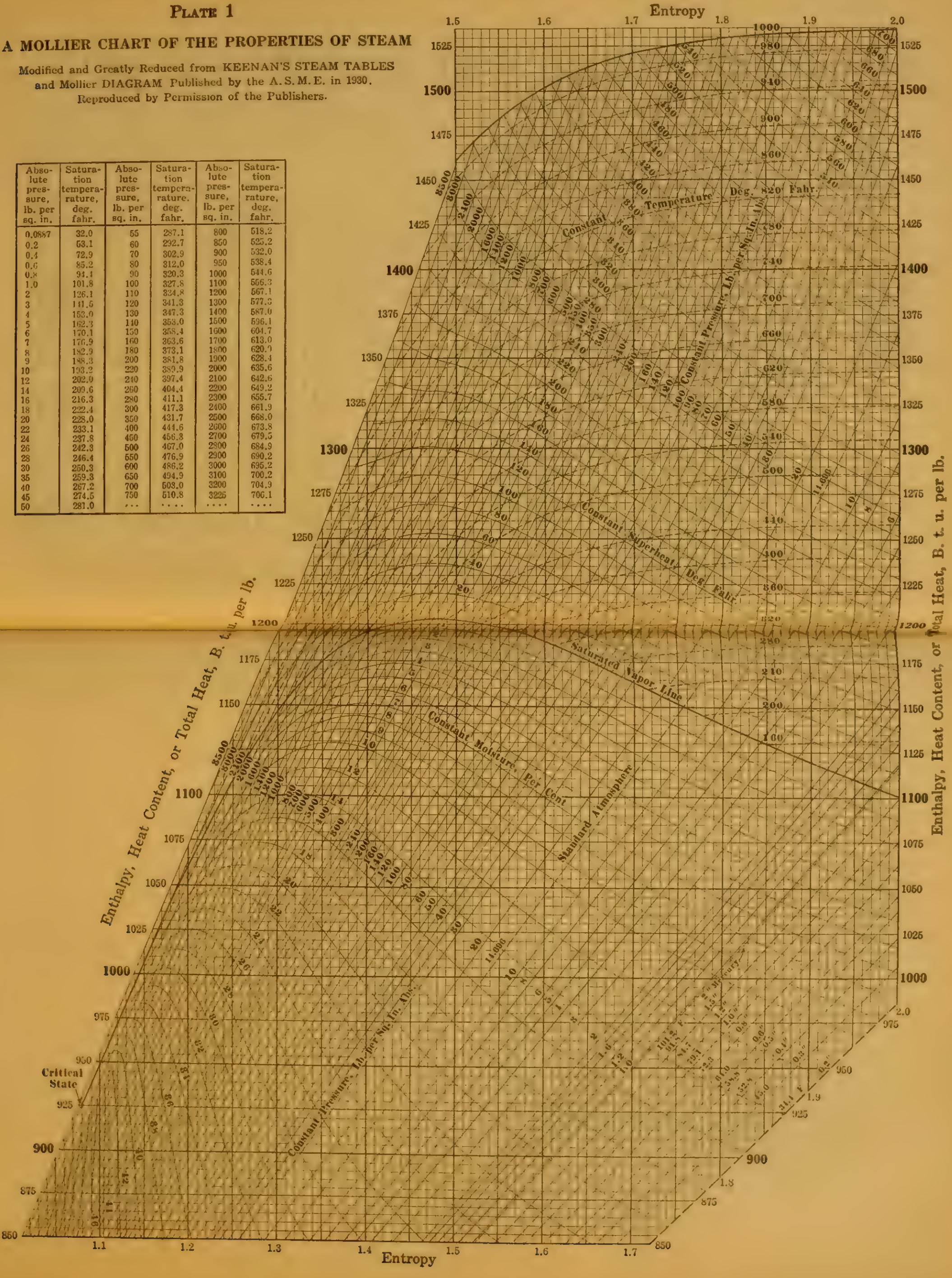
Abs. Press., Lb. per Sq. In. (Sat. Temp.)	Symbol	Temperature, Degrees Fahrenheit							
		650	700	750	800	850	900	950	1000
1600 (604.74)	$\overline{V}$	0.2999	0.3384	0.3708	0.4002	0.4264	0.4503	0.4725	0.4935
	h	1228.3	1278.6	1320.3	1360.0	1398.0	1434.3	1469.3	1502.9
	ϕ	1.3870	1.4313	1.4666	1.4986	1.5283	1.5555	1.5808	1.6042
1700 (612.98)	$\overline{V}$	0.2730	0.3119	0.3437	0.3721	0.3974	0.4204	0.4416	0.4616
	h	1215.5	1269.8	1313.4	1354.4	1393.8	1431.1	1466.8	1500.8
	ϕ	1.3706	1.4185	1.4553	1.4885	1.5250	1.5472	1.5730	1.5968
1800 (620.86)	$\overline{V}$	0.2483	0.2880	0.3195	0.3472	0.3716	0.3938	0.4140	0.4330
	h	1201.5	1260.6	1306.2	1348.7	1389.4	1427.8	1464.3	1498.8
	ϕ	1.3537	1.4058	1.4444	1.4787	1.5104	1.5392	1.5656	1.5897
1900 (628.39)	$\overline{V}$	0.2257	0.2665	0.2977	0.3248	0.3486	0.3700	0.3895	0.4076
	h	1186.0	1250.9	1298.9	1342.9	1385.0	1424.5	1461.7	1496.7
	ϕ	1.3358	1.3931	1.4336	1.4692	1.5019	1.5316	1.5585	1.5830
2000 (635.61)	$\overline{V}$	0.2044	0.2468	0.2781	0.3047	0.3289	0.3486	0.3673	0.3847
	h	1168.9	1240.7	1291.4	1337.0	1380.4	1421.1	1459.2	1494.7
	ϕ	1.3168	1.3802	1.4230	1.4599	1.4957	1.5242	1.5518	1.5765
2200 (649.25)	$\overline{V}$	0.1634	0.2120	0.2438	0.2696	0.2919	0.3115	0.3289	0.3451
	h	1125.8	1218.4	1275.7	1324.8	1371.2	1414.2	1454.0	1490.4
	ϕ	1.2719	1.3536	1.4021	1.4418	1.4779	1.5102	1.5390	1.5644
2400 (661.94)	$\overline{V}$	....	0.1816	0.2148	0.2401	0.2617	0.2805	0.2969	0.3121
	h	....	1192.8	1259.1	1312.3	1361.6	1407.1	1448.7	1486.1
	ϕ	....	1.3253	1.3813	1.4244	1.4628	1.4969	1.5270	1.5531
2600 (673.82)	$\overline{V}$	....	0.1540	0.1896	0.2151	0.2361	0.2542	0.2699	0.2842
	h	....	1162.5	1241.0	1299.2	1351.8	1399.8	1443.2	1481.7
	ϕ	....	1.2938	1.3602	1.4074	1.4483	1.4842	1.5157	1.5426
2800 (684.91)	$\overline{V}$	....	0.1275	0.1674	0.1934	0.2141	0.2315	0.2466	0.2603
	h	....	1123.8	1221.3	1285.5	1341.6	1392.1	1437.0	1477.3
	ϕ	....	1.2559	1.3384	1.3905	1.4342	1.4720	1.5048	1.5326
3000 (695.25)	$\overline{V}$	....	0.0983	0.1476	0.1742	0.1947	0.2118	0.2265	0.2396
	h	....	1066.3	1199.3	1271.1	1331.0	1384.3	1431.6	1472.9
	ϕ	....	1.2028	1.3155	1.3737	1.4203	1.4602	1.4945	1.5233
3200 (704.91)	$\overline{V}$	....	....	0.1293	0.1572	0.1777	0.1945	0.2059	0.2214
	h	....	....	1174.5	1255.9	1320.1	1376.1	1425.5	1468.4
	ϕ	....	....	1.2907	1.3567	1.4067	1.4487	1.4844	1.5144
3226 (706.10) Critical	$\overline{V}$	....	....	0.1271	0.1552	0.1757	0.1924	0.2067	0.2192
	h	....	....	1171.2	1253.8	1318.7	1375.0	1424.7	1467.8
	ϕ	....	....	1.2874	1.3545	1.4050	1.4472	1.4831	1.5133
3500	$\overline{V}$	....	....	0.1042	0.1349	0.1556	0.1720	0.1860	0.1981
	h	....	....	1129.7	1231.2	1303.0	1363.3	1415.8	1461.6
	ϕ	....	....	1.2484	1.3307	1.3866	1.4318	1.4698	1.5017

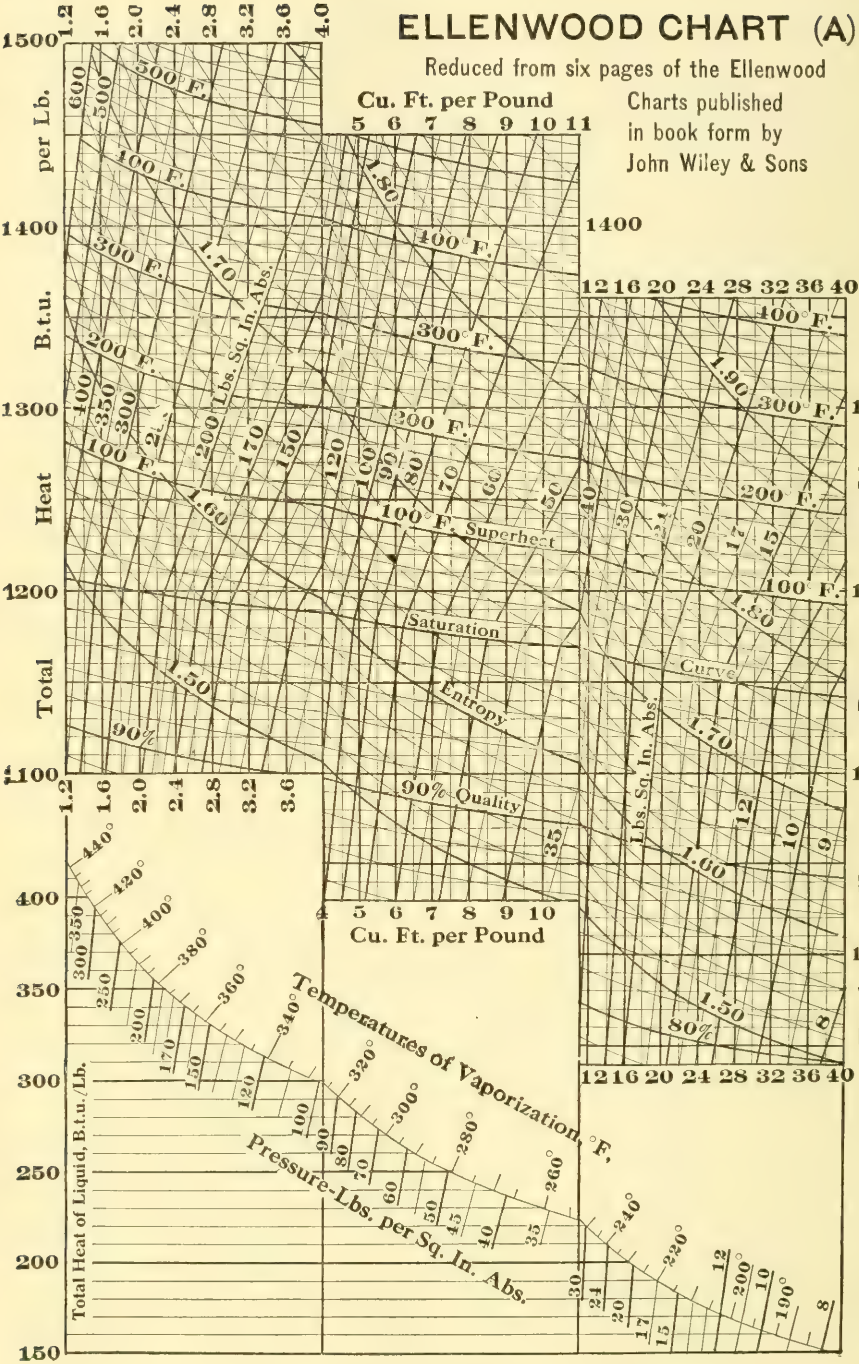
PLATE 1

A MOLLIER CHART OF THE PROPERTIES OF STEAM

Modified and Greatly Reduced from KEENAN'S STEAM TABLES
and Mollier DIAGRAM Published by the A. S. M. E. in 1930.
Reproduced by Permission of the Publishers.

Absolute pressure, lb. per sq. in.	Saturation temperature, deg. fahr.	Absolute pressure, lb. per sq. in.	Saturation temperature. deg. fahr.	Absolute pressure, lb. per sq. in.	Saturation temperature, deg. fahr.
0.0887	32.0	55	287.1	800	518.2
0.2	53.1	60	292.7	850	525.2
0.4	72.9	70	302.9	900	532.0
0.6	85.2	80	312.0	950	538.4
0.8	94.4	90	320.3	1000	544.6
1.0	101.8	100	327.8	1100	556.3
2	126.1	110	334.8	1200	567.1
3	141.5	120	341.3	1300	577.3
4	153.0	130	347.3	1400	587.0
5	162.3	140	353.0	1500	596.1
6	170.1	150	358.4	1600	604.7
7	176.9	160	363.6	1700	613.0
8	182.9	180	373.1	1800	620.9
9	188.3	200	381.8	1900	628.4
10	193.2	220	389.9	2000	635.6
12	202.0	240	397.4	2100	642.6
14	209.6	260	404.4	2200	649.2
16	216.3	280	411.1	2300	655.7
18	222.4	300	417.3	2400	661.9
20	228.0	350	431.7	2500	668.0
22	233.1	400	444.6	2600	673.8
24	237.8	450	456.8	2700	679.5
26	242.3	500	467.0	2800	684.9
28	246.4	550	476.9	2900	690.2
30	250.3	600	486.2	3000	695.2
35	259.3	650	494.9	3100	700.2
40	267.2	700	503.0	3200	704.9
45	274.5	750	510.8	3225	706.1
50	281.0	. . .	. . . .	. . . .	. . . .

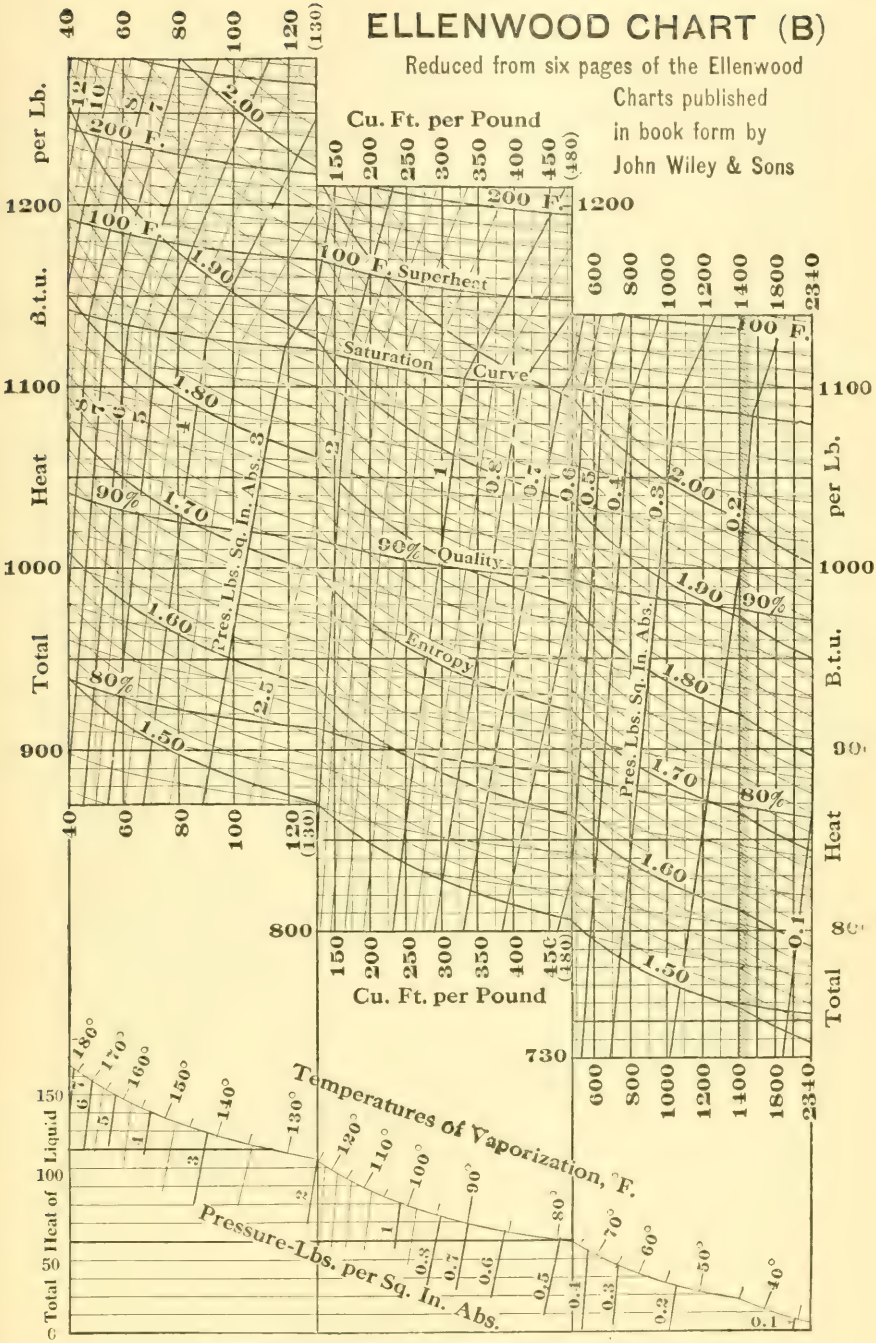




ELLENWOOD CHART (A)

Reduced from six pages of the Ellenwood Charts published in book form by John Wiley & Sons

Plate IIA. The Ellenwood Chart.



ELLENWOOD CHART (B)

Reduced from six pages of the Ellenwood Charts published in book form by John Wiley & Sons

Plate IIB. The Ellenwood Chart (Cont.)

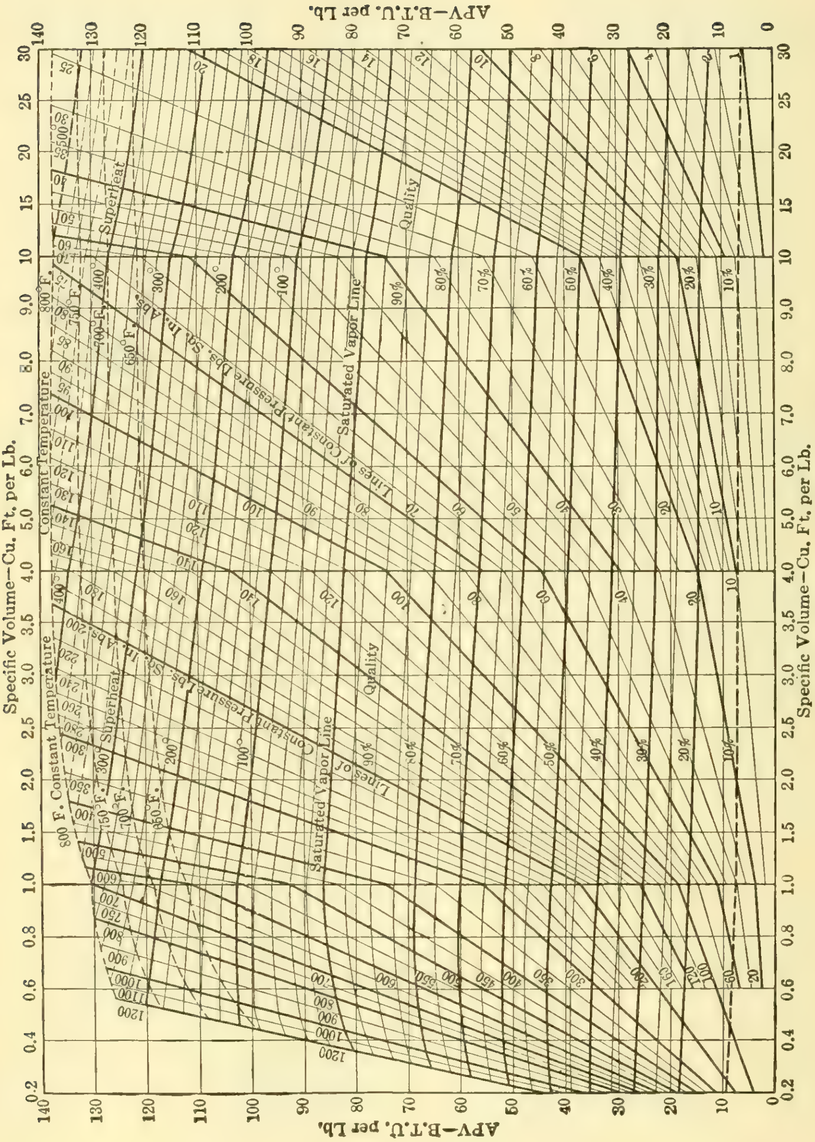


Plate III. Ellenwood APV-V Chart.

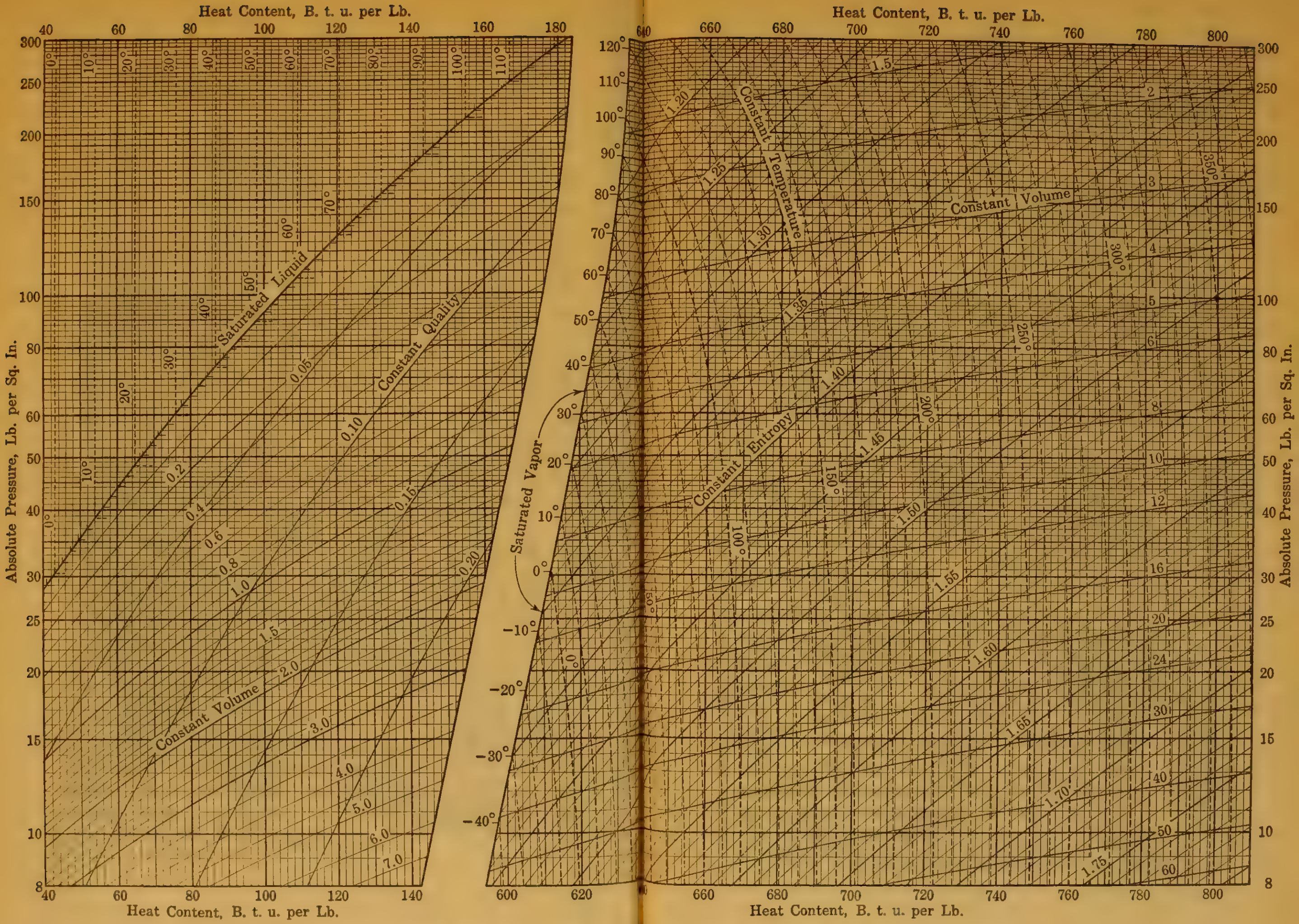


Plate IV.—A Mollier Chart for Ammonia, Modified from the one in Circular 142, U. S. Bureau of Standards.

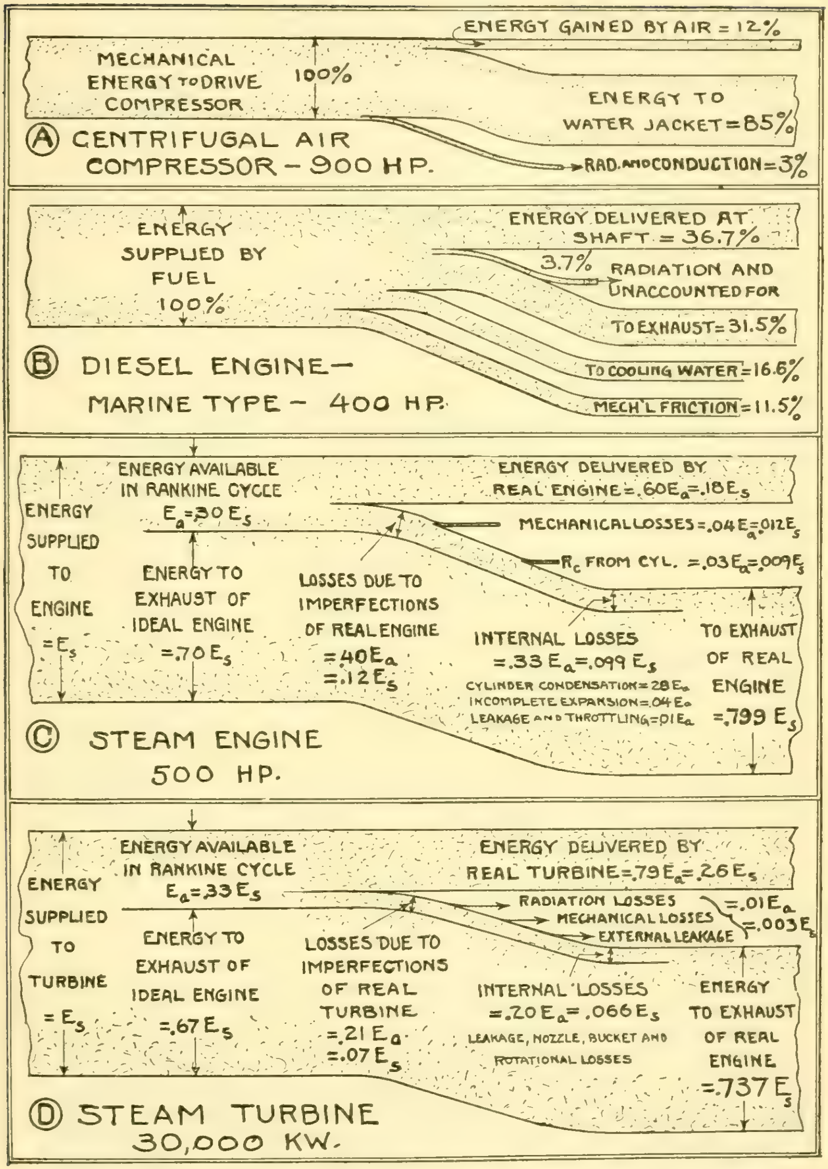
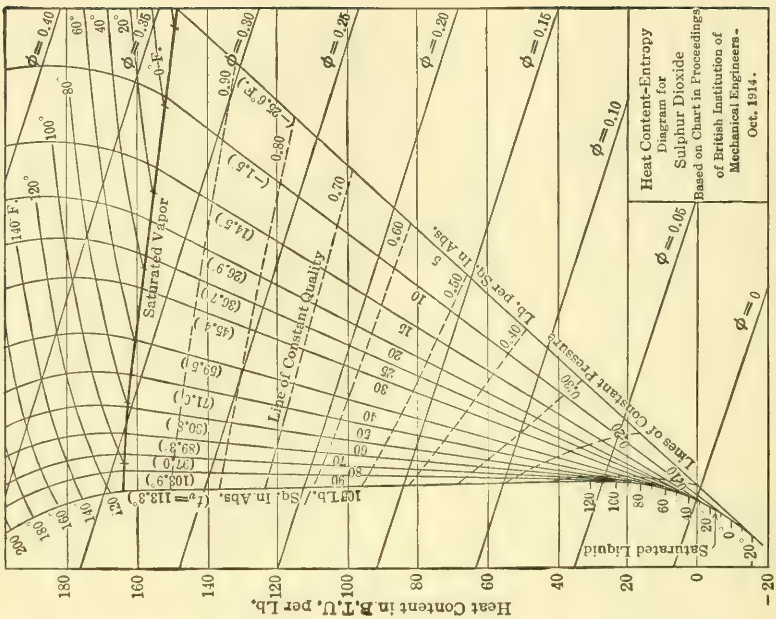
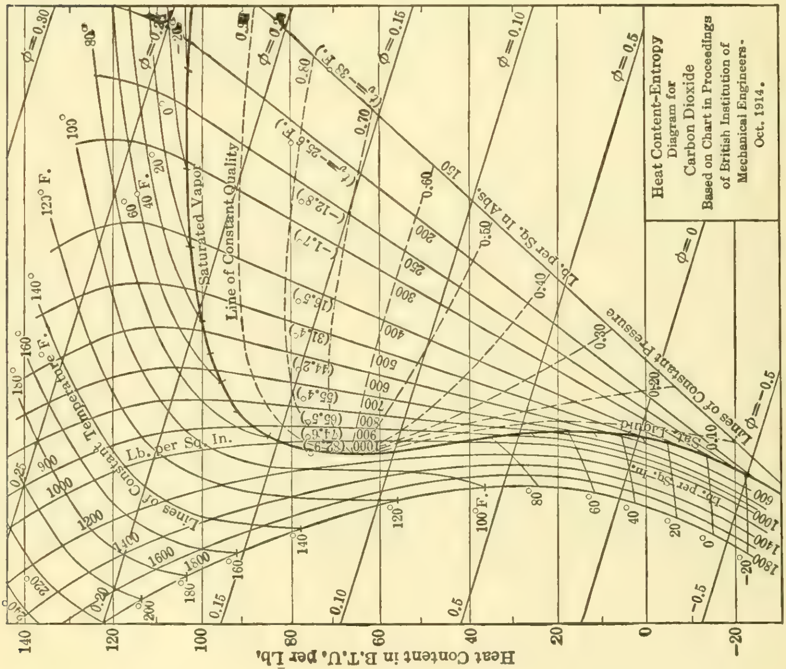

ENERGY GAINED BY AIR = 12%
MECHANICAL ENERGY TO DRIVE COMPRESSOR
100%
ENERGY TO WATER JACKET = 85%
RAD. AND CONDUCTION = 3%
(A) CENTRIFUGAL AIR COMPRESSOR – 900 HP.
ENERGY SUPPLIED BY FUEL 100%
ENERGY DELIVERED AT SHAFT = 36.7%
3.7% RADIATION AND UNACCOUNTED FOR
TO EXHAUST = 31.5%
TO COOLING WATER = 16.6%
MECH'L FRICTION = 11.5%
(B) DIESEL ENGINE – MARINE TYPE – 400 HP.
ENERGY AVAILABLE IN RANKINE CYCLE $E_a = .30 E_s$
ENERGY DELIVERED BY REAL ENGINE = $.60E_a = .18E_s$
MECHANICAL LOSSES = $.04E_a = .012E_s$
R_c FROM CYL. = $.03E_a = .009E_s$
ENERGY SUPPLIED TO ENGINE = E_s
ENERGY TO EXHAUST OF IDEAL ENGINE = $.70E_s$
LOSSES DUE TO IMPERFECTIONS OF REAL ENGINE = $.40E_a$ = $.12E_s$
INTERNAL LOSSES = $.33E_a = .099E_s$
CYLINDER CONDENSATION = $.28E_a$
INCOMPLETE EXPANSION = $.04E_a$
LEAKAGE AND THROTTLING = $.01E_a$
TO EXHAUST OF REAL ENGINE = $.799E_s$
(C) STEAM ENGINE 500 HP.
ENERGY AVAILABLE IN RANKINE CYCLE $E_a = .33E_s$
ENERGY DELIVERED BY REAL TURBINE = $.79E_a = .26E_s$
RADIATION LOSSES
MECHANICAL LOSSES
EXTERNAL LEAKAGE
= $.01E_a$ = $.003E_s$
ENERGY SUPPLIED TO TURBINE = E_s
ENERGY TO EXHAUST OF IDEAL ENGINE = $.67E_s$
LOSSES DUE TO IMPERFECTIONS OF REAL TURBINE = $.21E_a$ = $.07E_s$
INTERNAL LOSSES = $.20E_a = .066E_s$
LEAKAGE, NOZZLE, BUCKET AND ROTATIONAL LOSSES
ENERGY TO EXHAUST OF REAL ENGINE = $.737E_s$
(D) STEAM TURBINE 30,000 KW.

PLATE V. ENERGY STREAMS TYPICAL OF LARGE MODERN MACHINES.



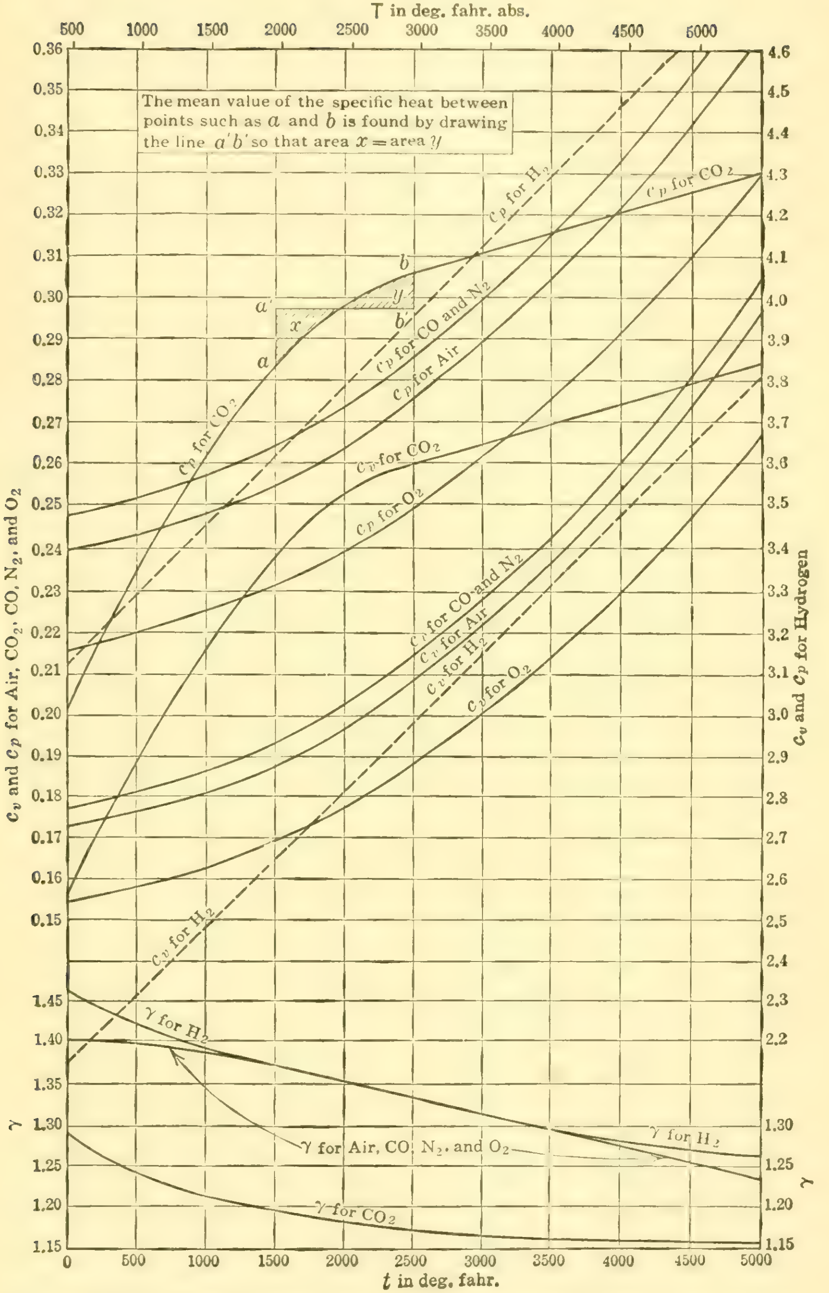
Heat Content-Entropy
Diagram for
Sulphur Dioxide
Based on Chart in Proceedings
of British Institution of
Mechanical Engineers -
Oct. 1914.

PLATE VII.



Heat Content-Entropy
Diagram for
Carbon Dioxide
Based on Chart in Proceedings
of British Institution of
Mechanical Engineers -
Oct. 1914.

PLATE VI.



The mean value of the specific heat between points such as a and b is found by drawing the line $a'b'$ so that area x = area y

PLATE VIII.—Chart Showing how the Instantaneous Values of c_p, c_v and y for Certain Gases Vary with the Temperature.

TABLE E.—INSTANTANEOUS SPECIFIC HEATS OF GASES

(In B.t.u. per pound per degree Fahrenheit)

$$C_p = M' + BT + CT^2$$

$$C_v = M + BT + CT^2$$

in which T = abs. temp. in deg. fahr., and the constants have the following values:

Gas	M'	M	B	C	For $T = 492$		For $T = 2000$	
					c_p	c_v	c_p	c_v
CO	0.2475	0.1765	0	$4.28\,(10)^{-9}$	0.2485	0.1775	0.265	0.194
H_2	2.98	2.00	$3.31\,(10)^{-4}$	0	3.140	2.160	3.64	2.66
N_2	0.2475	0.1765	0	$4.28\,(10)^{-9}$	0.2485	0.1775	0.265	0.194
O_2	0.216	0.154	0	$3.75\,(10)^{-9}$	0.217	0.155	0.231	0.169
Air	0.239	0.1705	0	$4.138\,(10)^{-9}$	0.240	0.1715	0.2555	0.187
$CO_2 (T < 2900)$	0.1625	0.1175	$8.86\,(10)^{-5}$	$-1.36\,(10)^{-8}$	0.2025	0.1575	0.285	0.240
$CO_2 (T > 2900)$	0.277	0.232	$9.55\,(10)^{-6}$	0				
C_2H_2	0.238	0.161	$3.11\,(10)^{-4}$	0	0.391	0.314	0.860	0.783
CH_4	0.216	0.092	$6.59\,(10)^{-4}$	0	0.540	0.416	1.53	1.41
C_2H_4	0.238	0.167	$2.42\,(10)^{-4}$	0	0.357	0.286	0.722	0.651

The values of M', M, B and C for air are those calculated in Part I, p. 86. For the other gases the values of these constants are based on the ones of Goodenough and Felbeck, as given on page 106, Bulletin No. 139, University of Illinois Experiment Station.

PROBLEMS

CHAPTER XXI

1. Trace the curves of velocity and pressure through the various types of steam turbines shown in Chart I (p. 2).

2. (*a*) What are the conditions to be fulfilled in order to have isentropic expansion of a fluid in a nozzle? (*b*) What is an ideal nozzle? (*c*) Explain the full significance of $\int VdP$ as applied to any fluid passing through an ideal nozzle.

3. Derive the expression for the maximum possible increase in the velocity energy of a fluid in passing through a thermally insulated nozzle, assuming uniform velocity at any transverse section.

4. Determine the velocity, in ft. per sec., and the temperature of a fluid issuing from an ideal horizontal nozzle, the pressure at entrance to, and discharge from, the nozzle being maintained at 45 and 15 lb. per sq. in. abs., respectively. The temperature of the fluid entering the nozzle is 100 F, and the entering velocity may be considered negligible. The fluid is (*a*) air, (*b*) hydrogen, (*c*) water.

Ans. (*a*) 1340, −51 F. (*b*) 5060, −64 F. (*c*) 66.9, 100 F.

5. Find the pressure, velocity, and area at the throat of the nozzle to give a discharge of 2000 pounds of fluid per hour for each case in problem 4.

Ans. (*a*) 23.8 lb. per sq. in., 1060 ft. per sec., 0.548 sq. in. (*b*) 23.3 lb. per sq. in., 4030 ft. per sec., 2.04 sq. in. (*c*) 15 lb. per sq. in., 66.9 ft. per sec., 0.0193 sq. in.

6. Superheated steam expands isentropically in an ideal nozzle from an initial state, in which the pressure is 300 lb. per sq. in. abs. and the superheat is 217 F, to a pressure of 100 lb. per sq. in. abs. The rate of flow is 1 lb. per sec. (*a*) Find the velocity of the steam, and the area of cross-section of the nozzle, at the sections where the pressures are 150, 160, 165, 170 and 180 lb. per sq. in. abs., respectively. (*b*) Determine the pressure, velocity, and cross-sectional area at the throat. (*c*) Determine the velocity and cross-sectional area at discharge.

Ans. (*a*) 1918, 1834, 1795, 1752, and 1658 ft. per sec.; 0.269, 0.265, 0.265, 0.266, and 0.269 sq. in. (*b*) 161 lb. per sq. in. abs., 1826 ft. per sec., 0.265 sq. in. (*c*) 2346 ft. per sec., 0.298 sq. in.

7. Dry saturated steam enters a circular nozzle at a pressure of 80 lb. per sq. in. abs. and at the rate of 3600 lb. per hour. (*a*) If this steam could expand *isentropically* and with uniform velocity at each cross-sectional area, find the velocity of the steam, and the diameter of the nozzle, at the sections where the pressures are 56, 50, 44, 40 and 15 lb. per sq. in. abs., respectively. (*b*) Determine the throat pressure and the throat diameter.

Ans. (*a*) 1192, 1357, 1522, 1634, and 2470 ft. per sec.; 1.072, 1.059, 1.053, 1.062, and 1.335 in. (*b*) 46.1 lb. per sq. in. abs.; 1.05 in.

8. (*a*) Compute the velocity of sound in dry air at a pressure of 15 lb. per sq. in. abs. and a temperature of 80 F. (*b*) If the above pressure and temperature exist at the throat of a nozzle in which air is expanding isentropically, determine the pressure and temperature of the air at entrance to this nozzle.

Ans. (*a*) 1140 ft. per sec. (*b*) 28.4 lb. per sq. in.; 188 F.

9. (*a*) Explain what factors determine the weight of fluid discharged by a nozzle. (*b*) What is meant by *under-expanding* and *over-expanding* nozzles? (*c*) Discuss the results given in Table XV. (*d*) Under what conditions may the empirical formulas for the flow through an orifice be used?

10. Explain what is meant by the following terms: (*a*) velocity coefficient of a nozzle, (*b*) actual discharge velocity, and (*c*) discharge coefficient.

11. What are the factors that reduce (*a*) the velocity coefficient, and (*b*) the discharge coefficient?

12. By test the actual discharge of air, from a square-edged orifice $\frac{3}{16}$ in. in diameter in a plate $\frac{1}{16}$ in. thick, was found to be 141 lb. per hr. when the pressures on the two sides of the orifice were maintained constant at 73.9 and 14.6 lb. per sq. in. abs., respectively, and the entering temperature of the air was 71 F. (*a*) Find the coefficient of discharge for this orifice under these conditions. (*b*) The same as (*a*) except the initial pressure and temperature were 114 lb. per sq. in. abs. and 60.4 F, and the weight actually discharged was 222 lb. per hr. (*c*) Discuss the effect of pressure ratio on the discharge coefficient of a square-edged orifice.

Ans. (*a*) 0.833. (*b*) 0.842.

13. In the equation, PV^n = a constant, discuss the values of n that give fairly satisfactory results for isentropic expansion of superheated steam, (*a*) when the pressure is very high, say above 2000 lb. per sq. in., and (*b*) when the pressure is fairly low, say less than 500 lb. per sq. in.

14. If the exit section of a nozzle is so large that the velocity is thereby reduced by the fraction x, determine what portion of the velocity energy is lost due to over-expansion in such a nozzle. Ans. $2x - x^2$.

15. The rate at which a fluid is discharged, in lb. per sec., from a thin plate orifice in a pipe may be well represented by $0.525\, C_d d^2\sqrt{(p_1 - p_2)\rho}$, if $(p_1 - p_2)/p_1$ is less than 1.5 per cent. C_d = coefficient of discharge, including the velocity of approach, as given by Fig. 263; d = diameter of orifice in inches; ρ = density of fluid at entrance to orifice, in lb. per cu. ft.; p_1 and p_2 = abs. press., in lb. per sq. in., at the up-stream and down-stream taps, respectively. (*a*) Find the weight of air, in lbs. per hr., discharged by a blower delivering to a pipe line having an inside diameter of 6 in. and containing a thin plate orifice 4 in. in diameter. The down-stream tap is 6 in. from the orifice; $p_1 = 15$; $t_1 = 60$ F; and $p_1 - p_2 = 2$ in. of water at 60 F. (*b*) The same as (*a*) except the orifice is 3 in. in diameter.

Ans. (*a*) 1610. (*b*) 810.

16. An air compressor delivers air into a pipe line that has an inside diameter of 3 in. This line contains a thin plate orifice 1.9 in. in diameter with a down-stream tap 6 in. from the orifice. The pressure and temperature at the up-stream tap are 100 lb. per sq. in. abs. and 150 F, respectively. If the orifice causes a pressure drop of 6 in. of water at 60 F find the rate, in pounds per hr., at which the air is flowing through the pipe. Ans. 1650.

17. (*a*) Explain what is meant by the term, *two-phase velocities*. (*b*) What is the effect of neglecting to consider two-phase velocities when calculating the discharge of a wet vapor?

18. (*a*) When is a vapor said to be supersaturated? (*b*) Define the degree of supersaturation. (*c*) What is the effect of supersaturation on the discharge from a nozzle?

19. Derive the expression for (*a*) the efficiency of a nozzle, and (*b*) the relation between nozzle efficiency and velocity coefficient when the entering velocity is negligible.

20. (*a*) Find the velocity, in ft. per sec., from a nozzle that has a velocity coefficient of 98 per cent when receiving steam having a pressure of 80 lb. per sq. in. abs., a temperature of 600 F, and a discharge pressure of 40 lb. per sq. in. abs., considering the entering velocity as negligible. (*b*) The same as (*a*) except the entering velocity is 300 ft./sec. (*c*) Find the nozzle efficiency for cases (*a*) and (*b*). (*d*) Find the heat content at the exit condition of the steam for cases (*a*) and (*b*). (*e*) Find the nozzle exit area for cases (*a*) and (*b*) for a flow of one pound per second.

Ans. (*a*) 1888. (*b*) 1910. (*c*) 96, 95.9 per cent. (*d*) 1255, 1255.1 B.t.u. per lb. (*e*) 1.014, 1.002 sq. in.

21. (*a*) What are some of the main considerations involved in deciding upon the *form* and *material* of a turbine nozzle? (*b*) What is a nozzle block?

22. (*a*) Explain the meaning of the terms *absolute* and *relative* velocities, as used in turbine calculations. (*b*) Explain how the actual drop in the heat content of a fluid in passing through a moving blade is related to the increase in the kinetic energy of the jet relative to the blade.

23. (*a*) Make a sketch of pure "impulse" steam turbine blades and the velocity diagram for the flow of steam through them assuming that there is no friction, turbulence, or heat transfer in passing through them. (*b*) The same as (*a*) except the turbulence and friction cause the relative velocity at exit to be 10 per cent less than at entrance. (*c*) If the same conditions as in (*b*) are assumed to hold when the entering relative velocity is 1400 ft. per sec., find the value of the heat content at exit relative to that at entrance to these blades.

Ans. 7.45 B.t.u. per lb. greater.

24. Make a sketch of pure "reaction" steam turbine blades and the velocity diagrams for the flow of steam through them assuming that there is a small loss in them due to friction and turbulence. (*b*) Derive the equation between the relative velocities and the actual entering and leaving heat contents.

25. (*a*) Steam enters a moving blade of a turbine with a relative velocity of 300 ft. per sec., a pressure of 70 lb. per sq. in. abs., and 60 deg. of superheat. If the steam leaves this blade with a relative velocity of 800 ft. per sec. and a pressure of 60 lb. per sq. in. abs., find its specific volume and entropy. (*b*) Determine the blade loss factor for this case. (*c*) If better blades were used so the loss factor is only 5 per cent what would the relative velocity at exit then become?

Ans. (*a*) 7.69 cu. ft. per lb., 1.673. (*b*) 15.4 per cent. (*c*) 841 ft. per sec.

26. Derive the general expression for the tangential force on a turbine blade, for assumed velocities, angles, and rate of flow.

27. For a given rate of flow, entering jet velocity, blade angles, and radius of rotation, show by means of vector diagrams and equations what value of the blade velocity will give the maximum torque.

28. Explain what is meant by *blade efficiency* and why it can never be equal to 100 per cent in any actual turbine.

29. Is maximum blade efficiency or maximum torque generally desired in a turbine? Why?

30. Steam enters the nozzles of a certain stage of an impulse turbine with a pressure of 80 lb. per sq. in. abs. and 80 degrees of superheat. The discharge pressure from these nozzles is 56 lb. per sq. in. abs., and the nozzle angle is 14 degrees. The pitch diameter of the buckets in this stage is 6 ft. and the entrance and exit angles of the buckets are equal. The turbine is designed to run at a normal speed of 1800 r.p.m. with 180,000 lb. of steam per hour passing through this stage. If the velocity coefficient for the nozzle is 97 per cent and the relative velocity at exit from the bucket is 90 per cent of that at entrance when operating at normal speed, find, (*a*) the bucket angles to give a minimum entrance loss at normal speed, (*b*) the absolute exit velocity, in magnitude and direction, (*c*) the horse power delivered to the buckets of this stage at normal speed, (*d*) the same as (*c*) by another method, (*e*) what per cent of the energy available in this stage is delivered to the buckets and explain why it differs from the diagram efficiency, (*f*) the torque, and (*g*) the torque by another method.

Ans. (*a*) 25° 45′. (*b*) 265 ft. per sec., 85°. (*c*) 1847. (*e*) 84.3. (*f*) 5360 lb. ft.

31. Assume the turbine of the previous problem to be blocked so that the shaft cannot rotate and that the steam pressures are such that under these circumstances the same nozzle velocity and steam flow are attained in this stage as in problem 30. Determine (*b*), (*c*), (*d*), (*e*), and (*f*). Draw the velocity diagram approximately to scale. Ans. (*b*) 1088 ft. per sec., 25° 45′. (*f*) 10,080 lb. ft.

32. (*a*) The same as problem 30, except the nozzle angle is 12 degrees. (*b*) Compare the results with those of problem 30.

Ans. (*a*) 22° 10′. (*b*) 226.6 ft. per sec., 87° 28′. (*c*) 1876. (*e*) 85.5. (*f*) 5475 lb. ft.

33. (*a*) The same as problem 31, except the nozzle angle is 12 degrees. (*b*) Compare the results with those of problem 31.

Ans. (*b*) 1088 ft. per sec., 22° 10′. (*f*) 10,200 lb. ft.

34. (*a*) Find the blade efficiency of the stage of the turbine specified in problems 30 and 32, for normal speed. (*b*) If the flow of steam through these blades could be without any friction or turbulence what would the blade efficiency then become? (*c*) What velocity ratio would give the maximum efficiency for the conditions given in (*b*), and what would this efficiency become?

Ans. (*a*) 89.3, 90.6 per cent. (*b*) 93.9 and 95.4 per cent. (*c*) 0.4852, 94.1 per cent.; 0.489, 95.7 per cent.

35. (*a*) Under what conditions does the blade efficiency of a reaction turbine become equal to $(v_1 \cos A - u + R_2 \cos D)u \div v_1^2$? (*b*) Why is it desirable to express this efficiency in terms of the angles A and D?

36. (*a*) Explain by the aid of a velocity diagram what is meant by velocity staging. (*b*) Explain what is meant by the Rateau or Zoelly type of multi-staging.

37. A steam turbine delivers 10,000 h.p. when using 108,000 pounds of steam per hour. The steam at the throttle has a pressure of 80 lb. per sq. in. abs., and a temperature of 400 F.; the exhaust pressure is 1.5 in. Hg abs. If this turbine has 8 stages, each of which has the same stage efficiency, and if the *mechanical losses* of the turbine amount to 1.5 per cent of the available energy, find for this turbine, (*a*) the engine efficiency, (*b*) the internal efficiency, (*c*) the reheat, (*d*) the reheat factor, (*e*) the stage efficiency, and (*f*) the specific entropy and volume of the steam at exhaust.

Ans. (*a*) 76.4 per cent. (*b*) 77.9 per cent. (*c*) 67.9 B.t.u. per lb. (*d*) 1.038. (*e*) 75 per cent. (*f*) 1.80, 397 cu. ft. per lb.

38. (*a*) Assuming the same data as in the previous problem except the turbine has 10 stages, what results will be affected and why? (*b*) Determine these new values. (*c*) What are the advantages and disadvantages of multi-staging a turbine?
Ans. (*b*) 1.04, 75.5 per cent.

39. (*a*) Name the various losses in a steam turbine and explain why each occurs. (*b*) Compare the relative magnitude of these losses.

40. (*a*) What is the effect of *reheat* in any one stage of a turbine assuming the pressures at entrance and exit to remain unaffected? (*b*) What is the effect of *reheat* in a combination of several stages in a turbine. (*c*) Define by words and also by an equation the term, *reheat factor*.

41. (*a*) Explain why some form of a steam chart is so valuable in making turbine calculations. (*b*) Explain the significance of the *condition curve* of steam in a turbine. (*c*) On some form of steam chart make a freehand sketch of the *condition curves* for two turbines, one of which has stage efficiencies appreciably higher than the other.

42. (*a*) Explain what determines the entrance angle of a turbine blade. (*b*) What determines the blade length? (*c*) Describe some form of blade construction. (*d*) What materials are used for turbine blades? (*e*) Discuss blade clearances.

43. Explain what is meant by the following terms: (*a*) operating speed, (*b*) critical speed, (*c*) static balance, and (*d*) dynamic balance.

44. Discuss, briefly, turbine vibration.

45. (*a*) Describe the small De Laval turbine that operates above its critical speed. (*b*) Describe a small turbine using velocity-compounding. (*c*) What is generally true regarding the first cost and thermal economy of small turbines?

46. Describe the general features of (*a*) medium-size turbines, and (*b*) large single-cylinder turbines.

47. In connection with turbines, explain the purpose of (*a*) synchronizing devices, (*b*) secondary valves, (*c*) safety trips, (*d*) packing glands, (*e*) diaphragms, and (*f*) auxiliary oil pumps.

48. Explain what is meant by the following turbine terms: (*a*) combination type, (*b*) opposed double-flow, (*c*) cross compound, (*d*) axial flow, and (*e*) radial flow.

49. (*a*) What is meant by a reheating turbine? (*b*) What are its advantages and disadvantages?

50. What are the special problems involved in using a turbine in marine work?

51. (*a*) For what purposes is steam often bled from a turbine? (*b*) Describe the pressure regulator used with a bleeder turbine.

52. What is a mixed-pressure turbine?

53. What are the chief troubles encountered with steam turbines?

CHAPTER XXII

1. (*a*) How are the first cost and thermal economy of a steam turbine affected by multistaging? (*b*) For what classes of service are steam turbines far superior to piston engines and vice versa? (*c*) Under what conditions may the steam rates of prime movers be justly taken as comparative measures of performance?

2. What is a reasonable value of the efficiency of a high grade 50,000 kw. electric generator running at 1800 r.p.m. and carrying (*a*) full load, (*b*) half load, and (*c*) one-quarter load?

3. The same question as the preceding one, except the generator is rated at 1000 kw. and operates at a speed of 3600 r.p.m.

4. Steam at a pressure of 260 lb. per sq. in. abs. and 140 deg. of superheat is generated in a hotel at a cost of *46 cents* per 1000 lb. This hotel keeps a 1000 kw., 3600 r.p.m., electrical generator operating continuously at full load by means of a steam prime mover exhausting at an absolute pressure of 17 lb. per sq. in. If the exhaust steam is credited with a value of — cents per 1000 lb. what is a reasonable estimate of the cost of the steam required to operate this prime mover one year if it is (*a*) a single cylinder, four-valve engine, (*b*) a compound, four-valve engine, and (*c*) a single stage turbine?

(The amount to be credited for the value of the exhaust steam is to be estimated by the student, assuming that there is sufficient use for this exhaust during the entire year.)

5. Determine the approximate cost of turbine-generators erected in the United States, assuming that the turbine is designed for a steam pressure of 400 lb. per sq. in. abs., a temperature of 700 F., and that the unit is rated at (*a*) 8000 kw. when operated condensing at 3600 r.p.m., and (*b*) 25,000 kw. when operated condensing at 1800 r.p.m.

6. For the unit giving test results as shown by the curves in Fig. 328, determine the engine efficiency of the turbine when the generator output is (*a*) 10,000 kw., and (*b*) 15,000 kw. Ans. (*a*) 75.2 per cent. (*b*) 76 per cent.

7. For the unit giving test results as shown by the curves in Fig. 330 determine the engine efficiency when the generator output is 15,000 and 23,900 kw., the exhaust pressure being (*a*) 0.5 in. Hg abs., (*b*) 1.0 in. Hg abs., and (*c*) 1.5 in. Hg abs.

Ans. (*a*) 69.8, 73.5 per cent. (*b*) 71.1, 78 per cent. (*c*) 71.7, 79.3 per cent.

8. Discuss the factors that cause the curves in Fig. 330 to be as they are.

9. Discuss the factors that cause typical engine efficiencies of turbine generators to vary with size as shown by Figs. 332, 333, and Table XVI.

10. Determine the engine efficiency of the turbine for which the test results are given in Table XVI and Fig. 332 for case (*a*) No. 6, (*b*) No. 7, (*c*) No. 9, and (*d*) No. 10.

11. Assuming the load on each unit of the previous problem to be 15,000 kw., determine for each case the probable value of (*a*) the combined engine efficiency, and (*b*) the engine efficiency of the turbine.

12. Using the data for the 1000 kw. bleeder turbine-generator unit as given in Fig. 334, for a load of 800 kw., determine: (*a*) the maximum amount of steam that could be extracted per hour and explain why this is the maximum, (*b*) the amount of steam going to the condenser per hour, (*c*) the combined engine efficiency of the unit, (*d*) the probable amount of energy per hour carried away from the turbine with the bled steam, and (*e*) the probable amount of heat absorbed per hour by the circulating water in the condenser.

Ans. (*a*) 30,000 lb. (*b*) 1000 lb. (*c*) 52.8 per cent. (*d*) 35,088,000 B.t.u. (*e*) 983,800 B.t.u.

13. For the mixed-pressure turbine, whose performance curves are given in Fig. 336, determine: (*a*) the steam used per kw-hr. when the load is 500 kw. and the turbine is using no by-product steam, (*b*) the same as (*a*) except the turbine is receiving 7500 lb. per hr. of by-product steam, (*c*) the power developed by the by-product steam and by the live steam when operating as in (*b*).

Ans. (*a*) 21 lb. (*b*) 28 lb. (*c*) 190; 310 kw.

14. If the throttle temperature and the exhaust pressure are kept constant, discuss the effect of increasing the throttle pressure of large regenerative units on the (*a*) ideal cycle efficiency, (*b*) engine efficiency, (*c*) thermal efficiency, and (*d*) space occupied per unit of power.

15. (*a*) For a very large single-cylinder multistage turbine with no steam extraction, the maximum amount of steam for which the last stage can be designed is limited by what? Why? (*b*) If this limit has been reached in the last stage could the maximum power obtainable from the turbine be increased by redesigning it to bleed as much steam as possible for feed water heating, assuming the same exhaust pressure and the same condition of the steam at the throttle as in (*a*)? Why? (*c*) What stages would be affected by the redesign involved in (*b*)? Why? (*d*) Explain what limits the amount of steam that may be bled from the turbine specified in (*b*).

16. (*a*) How will the thermal economy, power output, and stage pressures of a regenerative impulse turbine be affected by opening the overload valves, assuming that the condenser capacity is sufficient to keep the exhaust pressure constant and that the turbine was designed to give its maximum thermal economy when extracting steam at four stages? Why? (*b*) After having opened the overload valves, how will the thermal economy, power output, and stage pressures be affected by closing the valves in all extraction steam lines? Why?

17. (*a*) Will the condenser equipment generally installed with a turbine be able to maintain constant exhaust pressure for large overloads? Why? (*b*) Under what conditions would it be desirable to operate a turbine with the maximum output obtainable from it? Why?

18. (*a*) Determine the temperature of the feed water leaving the last heater of a large regenerative unit to give the minimum energy consumption under the following conditions: 4 heaters; throttle pressure, 1200 lb. per sq. in. abs.; throttle temperature, 800 F; exhaust pressure, 1 in. Hg abs.; stage efficiencies, 0.845 with superheated steam and $0.845x$ with a quality x; last stage leaving loss, 10 B.t.u. per lb.; mechanical-electrical efficiency of the unit, 94 per cent; pressure drop to heaters, 5 per cent; and terminal temperature difference in the last heater, 7 F. (*b*) Determine the energy consumption of this unit, in B.t.u. per kw-hr.

Ans. (*a*) 460 F. (*b*) 9350.

19. The same as problem 18, except 3 heaters are to be used.

Ans. (*a*) 421 F. (*b*) 9460.

20. The same as problem 18, except the throttle pressure is to be 1800 lb. per sq. in. abs.

Ans. (*a*) 502 F. (*b*) 9060.

21. The same as problem 18, except the throttle temperature is to be 900 F.

Ans. (*a*) 460 F. (*b*) 9000.

22. Discuss the factors involved in selecting the proper number of regenerative feedwater heaters to install in any specific case.

23. Using the test data as given in Fig. 340, calculate the total amount of steam bled, in lb. per hr., for loads of (*a*) 14,000 kw. and (*b*) 6000 kw.

Ans. (*a*) 29,400. (*b*) 12,500.

24. Determine the error involved by using Figs. 337, 338, 339 and Eq. (373), in estimating the energy consumption of the turbine for which the test results are shown in (*a*) Fig. 341 for a load of 44,000 kw., and (*b*) Fig. 342 for a load of 45,000 kw.

Ans. (*a*) −7.1 per cent. (*b*) −0.9 per cent.

25. (*a*) What are the most probable causes of the differences found in the previous problem? (*b*) What are the advantages of estimating the energy consumption as in the previous problem? (*c*) How could this energy be calculated more accurately? What disadvantage has such a method? (*d*) When would it be desirable to use one method in preference to the other? Why?

26. Explain why the steam pressure in the regenerative heaters varies with the load on the turbine.

27. Explain why the energy consumption curve in Fig. 342 has the changes in curvature as shown.

28. If the stage efficiency of a turbine is 84.5 per cent with superheated steam and $0.845x$ with a quality x, find the minimum exhaust pressure, in lb. per sq. in. abs., that could be used with this turbine without reheating, if the moisture in the turbine is not to exceed 10 per cent. The throttle pressure is 400 lb. per sq. in. abs. and the throttle temperature is (*a*) 700, (*b*) 800, and (*c*) 1000 F.

Ans. (*a*) 2. (*b*) 0.8. (*c*) 0.3.

29. The same as problem 28, except the pressure is 1200 lb. per sq. in. abs.

Ans. (*a*) 40. (*b*) 10. (*c*) 1.3.

30. (*a*) What are the serious results of using very wet steam in a turbine? (*b*) Name three methods that may be employed to reduce the percentage of moisture in the low-pressure stages of a turbine. (*c*) Discuss the relative merits of each method given in (*b*).

31. (*a*) What determines the maximum steam temperature that may be used in a steam plant? Why? (*b*) Why is a high steam temperature desirable?

32. (*a*) Discuss the factors involved in the selection of the steam pressure to be used in a large plant. (*b*) Discuss, briefly, the development of high-pressure steam plants.

33. (*a*) Explain the significance of the broken lines shown in Fig. 343. (*b*) What are the assumptions on which these condition curves are based? (*c*) Show by an example the application of these curves to a turbine using no reheating. (*d*) Show by an example how to apply these curves to a turbine that reheats the steam to the throttle temperature and also has a 10 per cent drop in pressure in the reheating system.

34. (*a*) Discuss the performance curves shown in Fig. 344. (*b*) For what conditions were these curves drawn? (*c*) If the throttle temperature were 900 F instead of 842, but the other conditions in the plant were unaltered, what changes would occur in the curves of this figure? Why?

35. (*a*) Explain how the size of a turbine-generator unit affects its thermal efficiency. (*b*) What other factors should be considered when comparing the advantages and disadvantages of one large unit in place of two or more smaller ones?

36. Explain what is meant by (*a*) gas reheating, (*b*) live-steam reheating, and (*c*) the combined system.

37. Show by an example the application of the curves in Fig. 345.

38. Show by an example the application of the curves in Fig. 346 (*a*) and (*b*).

39. At what pressure, approximately, should gas reheating occur in order to attain (*a*) the best thermal efficiency of the turbine, and (*b*) the best economic results from the station?

40. A steam plant has a turbine receiving steam at a throttle pressure of 1200 lb.

per sq. in. abs. and exhausting at a pressure of 1 in. Hg abs. When operating under its most economical load, 50,000 kw., the boiler feed pump and hot well pump are required to handle 8.5 lb. of water per kw-hr. output of the main unit. The hot well pump discharges against a pressure of 60 lb. per sq. in. abs., required mainly to overcome the fluid friction of the water through various heaters before delivery to the surge tank. Assume that this pump has an efficiency of 60 per cent and that it is driven by an electric motor having an efficiency of 85 per cent. From the surge tank the water enters the boiler feed pump at a temperature of 200 F and a pressure of 15 lb. per sq. in. abs. The resistance encountered by the feed water in passing through the feedwater heaters, the pipe line, and the regulating valve amounts to 150 lb. per sq. in., and the drop in steam pressure in passing through the superheater and main steam line is 170 lb. per sq. in. The efficiency of the boiler feed pump and its motor may be taken as 80 and 90 per cent, respectively. (*a*) Determine the power in kw., required to operate the motor driving the hot well pump at the specified load. (*b*) Find the power required to operate the feed pump motor. (*c*) What fraction of the output of the main unit is represented by the sum of the results in (*a*) and (*b*)? (*d*) Explain what becomes of the energy supplied to the motors that operate these pumps. Ans. (*a*) 43. (*b*) 800. (*c*) 1.69 per cent.

41. Suppose that the turbine of the previous problem is operating on the reheating-regenerative cycle under such conditions that its energy consumption at its most economical load is 9500 B.t.u. per kw-hr. Also assume that the auxiliaries are all electrically driven and that during one year the station operates under the following conditions: the operating factor is 90 per cent; the efficiency of the boiler unit is 84 per cent; the feed pump equipment is operated as in the previous problem and its recovery ratio is 60 per cent; the fuel used is coal with a heating value of 12,000 B.t.u. per lb., and requiring 45 kw-hr. per ton for the operation of all the fuel burning and handling equipment; the circulating water absorbs 8 B.t.u. per lb., and is pumped against a total head of 16 ft. with a pump having an efficiency of 75 per cent, and a motor efficiency of 90 per cent; the mechanical-electrical efficiency of the turbine-generator is 94 per cent; the transformer and cable losses in the station are 5 per cent; and the energy used by miscellaneous auxiliaries is negligible. (*a*) What per cent of the gross generator output is absorbed from the switchboard to operate all of the station auxiliaries? (*b*) Find the heat rate of the station in B.t.u. per kw-hr. (*c*) Determine the tons of coal required per year if the average load is 40,000 kw. (*d*) Calculate the amount of money that might be saved annually on fuel costing $5.00 per ton if the efficiency of the boiler unit could be increased to 86 per cent.

Ans. (*a*) 4.94. (*b*) 13,100. (*c*) 191,500. (*d*) $22,720.

42. (*a*) If the auxiliaries in the station of the previous problem could be replaced by other equipment that would require 20 per cent less energy from the switchboard during the year, how much could be saved in the cost of fuel required per year? (*b*) What reduction in the auxiliary energy consumption would become necessary to equal the fuel saving produced by the change in boiler efficiency specified in (*d*) of the previous problem? Which of these would probably be the easier to accomplish? Why? Ans. (*a*) $9850; (*b*) 46.8 per cent.

43. For the operating conditions as given in Fig. 349, for the Rankine cycle, find the quality of the steam at exhaust, and the station heat rate, in B.t.u. per kw-hr., if the throttle pressure is 400 lb. per sq. in. abs. and the throttle temperature is (*a*) 700, (*b*) 800, and (*c*) 900 F.

Ans. (*a*) 0.875; 16,050. (*b*) 0.895; 15,350. (*c*) 0.915; 14,750.

44. The same as problem 43, except the pressure is 800 and the cycle is the regenerative one. Ans. (*a*) 0.84; 13,700. (*b*) 0.86; 13,100. (*c*) 0.88; 12 600.

45. (*a*) For the reheating-regenerative cycle with one gas reheater and the conditions as given in Fig. 349 determine the percentage saving in fuel required by a station using steam at a temperature of 900 instead of 700 F., if the throttle pressure is 1400 lb. per sq. in. abs. (*b*) This difference means how much annual saving in the cost of fuel at $5 per ton in a station operating with an average load of 300,000 kw., the high heating value of the fuel being 12,000 B.t.u. per lb.

Ans. (*a*) 9.2 per cent. (*b*) $634,500.

46. How and why does an increase in the throttle pressure affect the amount of energy required by (*a*) the boiler feed pump, (*b*) the circulating pump, and (*c*) the fuel burning and handling equipment?

47. (*a*) Referring to Fig. 349 (*a*) and (*b*), explain why the energy consumption curves of the turbine-generator tend to reach a minimum value at some definite pressure. (*b*) Explain why the corresponding station heat rate curves attain their minimum values at a lower pressure and then rise more abruptly after this pressure is exceeded.

48. (*a*) In a large station using 4 closed feedwater heaters, a throttle pressure of 1400 lb. per sq. in. abs., and a temperature of 800 F, what feed water temperature would probably give the best thermal efficiency of the station? (*b*) Would this result be affected by changing the throttle pressure? Why? Ans. (*a*) 442 F.

49. Using the data as given in Fig. 351, determine what percentage of the throttle flow is represented by (*a*) the make-up coming from the evaporator, (*b*) the live steam going to the reheaters, (*c*) the steam coming from the superheater, and (*d*) the steam going to the condensers. Ans. (*a*) 4.13. (*b*) 9. (*c*) 109. (*d*) 73.5.

50. How does the feedwater temperature leaving the last heater, as given by the data in Fig. 351, compare with that obtained by using the curves in Fig. 350 for the same throttle pressure?

51. (*a*) Define the term, "*half-hour yearly load factor.*" (*b*) Define the term "*annual capacity factor,*" and give the other names that are also used for it.

52. (*a*) If a central station contains five 60,000 kw. turbine-generators and during one year delivers 1.5 billion kw-hr., find its annual use factor. (*b*) If the maximum half-hour load carried by this station is 250,000 kw. during a year, find its half-hour yearly load factor. (*c*) If this station cost $100 per kw. and the fixed charges are 15 per cent, determine the amount of these charges per kw-hr. sent out. (*d*) If the average annual heat rate of this station is 13,500 B.t.u. per kw-hr. when using coal costing $2 per ton and having a heating value of 10,000 B.t.u. per lb., find the fuel cost per kw-hr. (*e*) If labor, supplies, and maintenance cost 0.15 cent per kw-hr., find the cost of a kw-hr. of energy delivered by this station.

Ans. (*a*) 57.1 per cent. (*b*) 68.5 per cent. (*c*) 3 mills. (*d*) 1.35 mills. (*e*) 5.85 mills.

53. Assume the same conditions as given in the previous problem except the cost of fuel. Determine the price that would have to be paid for the fuel to make the fuel cost just equal to the fixed charges per unit of energy output.

Ans. $4.44 per ton.

54. Suppose that by eliminating the building and by installing larger units a 500,000 kw. station could be built for $60 per kw., and that this station would operate with an average annual heat rate of 13,000 B.t.u. per kw-hr. when using coal having a heating value of 11,000 B.t.u. per lb. Assuming fixed charges as 15 per cent, also labor, supplies, and maintenance as 0.15 cent per kw-hr., determine the cost of energy from this station for annual capacity factors of 25, 50, and 75 per cent, if the cost of fuel is (*a*) $2 per ton, (*b*) $4 per ton, and (*c*) $6 per ton.

Ans. (*a*) 6.79, 4.74 and 4.05 mills.

55. The same as the previous problem except the outdoor station is a mercury steam one assumed to have an average annual heat rate of 10,000 B.t.u. per kw-hr., and assumed to cost $75 per kw. Ans. (*a*) 7.55, 4.98 and 4.12 mills.

CHAPTER XXIII

1. Summarize briefly, the main points that must be considered in selecting a steam prime mover for a given class of work.

2. Explain how the following items are factors in the selection of a reciprocating steam engine: (*a*) rotative speed, (*b*) piston speed, (*c*) steam pressure, (*d*) steam temperature, and (*e*) type of engine.

3. (*a*) Make a sketch of a simple D-valve. (*b*) What are its chief advantages and disadvantages? (*c*) For what class of work is it suitable?

4. The same as problem 3, except the Sweet type of valve is used.

5. Describe a Corliss valve. Give its advantages and disadvantages.

6. (*a*) What is a double-beat type of poppet valve? (*b*) What are its advantages and disadvantages?

7. What are the advantages and disadvantages of making the piston function as an exhaust valve?

8. For what class of service is a throttling engine suitable? Why?

9. (*a*) What are *high-speed single-valve engines*? (*b*) What are their usual ranges of power and speed?

10. Compare the high-speed Corliss with the H.S.S.V. engine.

11. (*a*) Describe the low-speed type of Corliss engine. (*b*) Explain why it cannot not be run at a high speed.

12. How does the first cost of a uniflow engine compare with that of other types of the same piston speed and power? Why?

13. Name a city with which you are familiar, and estimate the extent to which all of its transportation needs are dependent upon heat engines.

14. (*a*) What type of heat engine predominates in the aeronautical field? Why? (*b*) In the automobile? Why? (*c*) On the railroad? Why?

15. What types of engines are used in pumping stations for pipe lines? Why?

16. Discuss, briefly, the development of the steam locomotive.

17. What are the reasons why large locomotives are not commonly driven by (*a*) steam turbines, or (*b*) Diesel engines?

18. (*a*) Name the kinds of heat engines that are used in the various classes of marine work. (*b*) Explain why each one is well suited for its class.

19. Discuss the relative merits of gears and electric drive when steam turbines are used to propel large vessels.

20. Describe the power plant of the U.S.S. *Saratoga*, and explain how the speed of the vessel is controlled.

21. Using the data given in Table XXI (p. 162) calculate the exponent that should be applied to the ratio of speeds to give the corresponding ratio of shaft horsepower as represented by: (*a*) Tests 1 and 2, (*b*) Tests 2 and 3, and (*c*) Tests 4 and 5.
Ans. (*a*) 5.61. (*b*) 3.32. (*c*) 2.96.

22. Using the data given in Table XXI calculate the saving in the cost of fuel required to run the *Saratoga* 3000 miles at a speed of 10.72 knots instead of 26.68 knots. Assume the fuel costs $0.90 per barrel of 336 lb. Ans. $13,840.

23. Determine the plant thermal efficiency of the *Saratoga* when operating as shown by test (*a*) No. 1, (*b*) No. 2, (*c*) No. 6.

24. Discuss the results of problems 21, 22, and 23.

CHAPTER XXIV

1. Explain what is meant by (*a*) pressure-responsive governors, and (*b*) speed-responsive governors.

2. Give examples of the applications of (*a*) and (*b*) of the previous problem.

3. Speed-responsive governors are subdivided into what classes?

4. Explain what is meant by (*a*) spindle governors, (*b*) shaft governors, and (*c*) hydrodynamic governors.

5. Name and explain the various methods of governing resulting from the application of the speed-responsive governors to (*a*) reciprocating steam engines, (*b*) impulse steam turbines, (*c*) reaction turbines, (*d*) Otto engines, and (*e*) Diesel engines.

6. If a steam engine has a speed of 300 r.p.m. at full load and a speed of 310 r.p.m. at no load, find the coefficient of regulation of the governor of this engine.
Ans. 3.28 per cent.

7. Explain what is meant by the following terms as applied to governors: (*a*) stability, (*b*) strength, and (*c*) sensitiveness.

8. (*a*) Make a sketch showing the essential elements of a simple flyball governor. (*b*) Make a vector analysis of all the forces actually acting on the balls of this governor when running at some definite rotative speed. (*c*) Derive the equation for the height of cone of this governor. (*d*) Make a vector analysis of all the forces that would have to act on the governor balls to hold them statically in the same position as in (*b*).

9. In a simple Watt governor, how is the height of cone affected by (*a*) the length of the weight arm, (*b*) the magnitude of the weight, (*c*) the method of suspension, and (*d*) the rotative speed?

10. In a simple pendulum governor the length of the weight arm from the pin to the center of gravity of the weight is 10 in. If θ be the angle between the weight arm and the vertical axis of rotation, determine the speed of this governor, in r.p.m., when θ is, (*a*) 40, (*b*) 50, and (*c*) 60 degrees.
Ans. (*a*) 67.8. (*b*) 74.0. (*c*) 83.9.

11. The same as problem 10, except the length of the arm is 15 in.
Ans. (*a*) 55.3. (*b*) 60.4. (*c*) 68.5.

12. The same as problem 10, except the length of the arm is 20 in.
Ans. (*a*) 47.9. (*b*) 52.3. (*c*) 59.3.

13. Increasing the speed of the simple pendulum governor has what effect on the (*a*) coefficient of regulation, and (*b*) strength of the governor?

14. Make a sketch of, and explain the function of, some form of damping device as applied to a governor.

15. The same as problem 8, except the governor is to be of the Porter type.

16. (*a*) In Fig. 374, what does the force C acting on the pin 2 actually represent when the governor is rotating? (*b*) What does this force represent when the governor is not rotating?

17. What advantages are gained by using the Porter instead of the Watt type of governor?

18. What would be the advantages and disadvantages of an isochronous governor?

19. In a Porter governor of the simple rhombic form, determine the height of the cone at a speed of 300 r.p.m. if the ratio of the weight of the central weight to that of each ball is (*a*) 5 : 1, (*b*) 10 : 1, and (*c*) 20 : 1.

Ans. (*a*) 2.35 in. (*b*) 4.31 in. (*c*) 8.22 in.

20. Determine the collar movement of the governor for each case in problem 19, neglecting all resistances to such movement, when the speed increases 3 per cent.

Ans. (*a*) 0.269 in. (*b*) 0.494 in. (*c*) 0.94 in.

21. If the governor of problem 19 has two balls each weighing 1 pound, find the actual governing force that results from an increase of speed of 3 per cent in each case. Ans. (*a*) 0.38 lb. (*b*) 0.72 lb. (*c*) 1.33 lb.

22. Explain by the aid of a diagram how the characteristic curves of a governor may be useful in analyzing governor performance.

23. (*a*) Name the various resistances that must be overcome by a governor. (*b*) Discuss the effect of these forces on the governor action.

24. What are the main advantages of the spring-balanced spindle governor?

25. (*a*) Make a sketch of a simple type of shaft governor. (*b*) Derive the relation between the tension of the spring, the magnitude of the weight, the radius of rotation, and the speed.

CHAPTER XXV

1. Define the following terms: (*a*) forward stroke, (*b*) back stroke, (*c*) dead center, (*d*) connecting rod, (*e*) eccentric rod, (*f*) running over, and (*g*) running under.

2. Explain fully the meaning and significance of the terms, finite length, infinite length, and angularity, as applied to the connecting rod and eccentric rod of an engine.

3. Sketch an eccentric, name its several parts, explain its function, and give the reason for its use. Define eccentric throw.

4. Sketch an external D-valve on its seat in central position for a double-acting engine, and define the following terms: (*a*) central position, (*b*) steam edge, (*c*) exhaust edge, (*d*) port width, (*e*) port length, (*f*) steam lap, (*g*) exhaust lap, and (*h*) negative lap.

5. Explain the meaning of the following terms, as applied to an external D-valve: (*a*) steam chest, (*b*) exhaust cavity, (*c*) valve travel, (*d*) valve displacement, (*e*) valve opening to steam, and (*f*) valve opening to exhaust.

6. What arrangement of engine and what position relative to the observer are assumed in the text in drawing conventional valve diagrams?

7. For an external D-valve, sketch a rectilinear diagram of valve displacements, head end only, with complete notation.

8. For an external D-valve, sketch a rectilinear diagram of valve displacements, crank end only, with complete notation.

9. Define *conjugate events.* What events are so related, and what consequences follow?

10. Define, and illustrate by sketch, lead and angle of advance. Why is lead employed? What is the position of the eccentric relative to the crank, for a direct-driven external valve?

11. Draw one or more rectilinear diagrams, with crank and eccentric added thereto in their correct relative positions for each of the four principal valve events, head end only, and by projecting therefrom show the development of the head end indicator diagram during one revolution of the crank.

12. State and prove the fundamental principle of the Bilgram diagram.

13. Sketch a Bilgram, Sweet, or Zeuner diagram for the head (or crank) end of an external D-valve, with positive (or negative) exhaust lap.

14. (*a*) Draw an elliptical diagram for the head (or crank) end of an external D-valve. (*b*) Show how this diagram may be derived from the Bilgram, Sweet, or Zeuner diagram.

15. Explain, with sketch, the effect of the angularity of the connecting rod. If the horizontal diameter of the crank circle represents the stroke of the piston, how may the true piston position for any crank position be located on it?

16. In nearly all slide valve engines, the *length of valve stem* is adjustable. Discuss for both head and crank ends the effect of shortening the valve stem, without other change, upon the (*a*) laps, (*b*) leads, and (*c*) principal valve events.

17. Given a plain external D-valve, if the *eccentric is advanced* (i.e., the angle of advance increased), what will be the effect upon the (*a*) travel of the valve, (*b*) leads, (*c*) maximum opening, (*d*) laps, and (*e*) time when the principal valve events occur, with reference to the position of the piston. This may be analyzed by using the Bilgram diagram.

18. Explain why early opening to steam (admission) is desirable, and how it is accomplished. What relation has it to the amount of compression and the speed of the engine? Sketch an indicator digram showing the effect of late admission.

19. (*a*) What is the main object of using compression in a steam engine? (*b*) Discuss the other advantages and the disadvantages that result from compression.

20. Why, and to what extent, is early release desirable? To what other event is release related? Can it always be secured as desired with a simple D-valve?

21. Explain, by reference to the Bilgram diagram, the disadvantages and limitations of the simple D-valve when used for early cutoffs.

22. Discuss valve friction, and some of the means employed in design for reducing it.

23. With the crank on head end dead center, show by a sketch the position of the eccentric (with $\alpha = 30°$) for driving a horizontal external valve, when the mean position of the eccentric rod is inclined 15° downward to the left, and a centrally pivoted reversing lever is interposed between this rod and the valve stem. Also show the rocker and valve in corresponding positions.

24. State the general characteristics of the "high-speed" type of engine, and show by sketch how the indicator diagram changes with variations in load.

25. State the reasons for employing a special form of valve instead of the simple flat D-valve in the "high-speed" type of engine.

26. On a sketch, with crank on head end dead center, (*a*) show the path of a swinging eccentric which is pivoted diametrically opposite the crank, locate three eccentric positions on this path, and label the throws and angles of advance of each; (*b*) draw the head end Bilgram diagram with the steam and exhaust lap circles in the three corresponding positions and for each find the crank positions for all events; and (*c*) by projection determine the corresponding three indicator diagrams. Label completely and explain.

27. Discuss the general characteristics of independent cutoff gears.

28. In what are known as the High-Speed Corliss or "four-valve" engines, how are the several valves driven, and how is governing effected? What advantages has this type over the single-valve high-speed type previously discussed?

29. Explain the action of the trip cutoff Corliss valve gear, and the effect of load changes upon the several valve events. What limitation does this gear impose upon the engine speed, and with the single eccentric gear what is the latest cutoff obtainable?

30. Discuss the object, arrangement, and functioning of the following gears: (*a*) Stephenson Link, (*b*) Marshall, (*c*) Joy, (*d*) Walschaert, (*e*) Baker, and (*f*) Poppet.

CHAPTER XXVI

1. Explain what is meant by heat conduction.

2. (*a*) Define the term, *specific thermal conductivity.* (*b*) Give the units in which this term is generally expressed in engineering work in this country, and also in the c.g.s. system. (*c*) Derive the relation between these units. (1 B.t.u. = 252 gram calories, and 1 ft. = 30.48 cm.)

3. At a temperature of 500 F, what is the approximate ratio of the specific thermal conductivity of (*a*) copper to aluminum, (*b*) copper to brass, (*c*) copper to soft steel, and (*d*) copper to 85 per cent magnesia?

4. Determine the total resistance to the conduction of heat through a 9-inch insulating wall made up of equal thicknesses of three materials having specific thermal conductivities of 0.5, 0.4, and 0.3 respectively. Ans. 23.5.

5. Discuss the relative importance of heat conduction through solids to the general problem of heat transmission in most engineering applications.

6. The fire tubes of a locomotive boiler have an outside diameter of 2 inches and a thickness of 0.1 in. If 6000 B.t.u. per hr. pass through each square foot of outside tube surface when the average temperature difference between the gas and steam is 1000 F, find the portion of this temperature drop that is due to the metal, (*a*) if mild steel ($k = 300$) is used, and (*b*) if copper tubes ($k = 2600$) are used.
Ans. (*a*) 0.21 per cent. (*b*) 0.024 per cent.

7. The brass tubes of a surface condenser have an outside diameter of 0.625 inch, and are 0.05 in. thick. If 9600 B.t.u. are being transmitted per hr. per sq. ft. of outer tube surface when the mean temperature difference between the steam and water is 15 F, and if the mean tube temperature is 85 F, find the temperature drop from the outer to the inner surface of the brass tube. Ans. 0.69 F.

8. The cross-section of a wall of a cold storage room consists of 16 in. of brick ($k = 3.6$), an air space equivalent to the resistance of about 8 in. of brick, 8 more in. of brick, 4 in. of corkboard ($k = 0.3$), and 1.5 in. of cement plaster ($k = 2.3$). (*a*) Find the quantity of heat transmitted through the wall by conduction per day per sq. ft. of wall surface, when the temperatures of the inner and outer wall surfaces are 30 and 70 F, respectively. (*b*) The same as (*a*) except the corkboard is left out. Ans. (*a*) 42 B.t.u. (*b*) 100 B.t.u.

9. A 6-in. steel pipe (O.D. = 6.625) is carrying superheated steam at a temperature of 500 F and the surrounding air temperature is 80 F. Consider 5 different cases with 5 different coverings of 85 per cent magnesia having thicknesses of 1, 2, 3, 4, and 5 in., respectively, and assume that the corresponding temperatures of the outer insulation surfaces are found to be 150, 122, 109, 103, and 100 F. Assume that the temperature of the *outer* pipe surface is that of the steam, and that the contact resistance between the pipe and the insulation may be neglected. (*a*) Find the quantity of heat transmitted by conduction through each thickness of insulation in B.t.u. per hr. per sq. ft. of outer *pipe* surface. (*b*) Plot the results of (*a*) against thickness of insulation.

10. With the same steam pipe as in problem 9 the heat lost from the *bare pipe* would be 1455 B.t.u. per hr. per sq. ft. of pipe surface. The fixed charges on the insulation, in dollars per year per sq. ft. of pipe covered, are $0.12 + 0.06y$, where y is the thickness of the insulation in inches. Assume the value of the energy in the steam main to be 35 cents per million B.t.u. (*a*) Determine the gross and net savings due to the insulation, in dollars per year per sq. ft. of pipe surface, for each case. (*b*) Plot a curve showing the results of (*a*) against thickness of insulation. (*c*) Determine the most economical thickness of insulation.

11. Heat is being transmitted entirely by conduction through a layer of stagnant gas ($k = 0.25$) 0.1 in. thick, thence through a 0.4 in. cast iron plate ($k = 320$), and a stagnant 0.1 in. water film ($k = 4.8$). The temperature at the surface of the water film farthest from the plate is 150 F, and 700 B.t.u. are being transmitted per hr. per sq. ft. of surface. Contact resistances are neglected. (*a*) Find the temperature drop in the gas film, in the cast iron, and in the water film. (*b*) Assuming straight-line temperature drops, plot temperature against distance through the three media. Ans. (*a*) 280, 0.875, 14.6 F.

12. Explain what is meant by: (*a*) free convection, (*b*) forced convection, (*c*) viscous flow, (*d*) critical velocity, and (*e*) absolute viscosity of a fluid.

13. The approximate values of the absolute viscosity, μ, in centipoises, is given in terms of the temperature, t, in deg. fahr., by the following equations:

(*a*) Air, $\mu = 0.0165 + 2.5\,(10)^{-5}\,t$
(*b*) Carbon dioxide, $\mu = 0.0133 + 2.5\,(10)^{-5}\,t$
(*c*) Nitrogen, $\mu = 0.0155 + 2.5\,(10)^{-5}\,t$
(*d*) Steam, $\mu = 0.0083 + 2.0\,(10)^{-5}\,t$

Determine the critical velocity, in ft. per sec., of each of these fluids at a temperature of 500 F and a pressure of one atmosphere when flowing through a steel tube having an inside diameter of 3 in. Ans. (*a*) 1.78. (*b*) 1.04. (*c*) 1.78. (*d*) 1.80.

14. The same as problem 13 except the temperature is 300 F.

15. (*a*) Explain what is meant by the term *film coefficient*, and name some of the most important facts that determine its value. (*b*) Explain why the film coefficient is so important in the study of heat transmission.

16. Show that each of the expressions in parenthesis in Eq. (413) is dimensionless when using the four fundamental units of (*a*) force (F), length (L), time (T), and temperature (θ); or (*b*) mass (M), length (L), time (T), and temperature (θ).

17. (*a*) Calculate the rate of *free convection* from each sq. ft. of surface of a bare horizontal pipe line having an outside diameter of 1.9 in. and a surface temperature of 100 F, when surrounded by still air at a temperature of 0 F. (*b*) Check the result of (*a*) by using Fig. 484. (*c*) Determine the equivalent value of the film coefficient.

Ans. (*a*) 98 B.t.u. per hr. per sq. ft.

18. Hot paraffin oil at a main body temperature of 250 F is flowing through the inner pipe of a heat exchanger with a velocity of 3.5 ft. per sec. The inside diameter of this pipe is 1.61 in. For this hot oil the specific heat, c, is 0.52; the specific thermal conductivity, k, is 1.0; the density, ρ, is 56; and the viscosity, μ, at the main body temperature is 2.1. If cold oil is flowing through the annular space surrounding the pipe, find the film coefficient of heat transfer for the oil being cooled.

Ans. 122 B.t.u. per hr. per sq. ft. per deg. fahr.

19. The film temperature of the circulating water in a new steam condenser is 80 F; the tubes are 9 ft. long and have an inside diameter of 0.9 in. Find the water film coefficient of heat transfer at water velocities of 2, 4, 6, 8, and 10 ft. per sec.

Ans. 395, 688, 951, 1197, 1430 B.t.u. per hr. per sq. ft. per deg. fahr.

20. (*a*) Explain what is meant by the term *mass velocity* in heat transmission (*b*) Does its value affect the gas film coefficient appreciably?

21. (*a*) Flue gas at a temperature of 600 F is flowing at right angles to rows of staggered tubes in a transverse flow-economizer. The tubes have an outside diameter of 4.5 in., and are placed on 8-in. centers. The mass velocity of gas flow is 0.8 lb. per sec. per sq. ft. of net section. If the temperature of the tube surface is 300 F, find the gas film coefficient of heat transfer. (*b*) Find the increase in the film coefficient if the mass velocity of gas flow is increased 25 per cent.

Ans. (*a*) 4.73 B.t.u. per hr. per sq. ft. per deg. fahr. (*b*) 17 per cent.

22. Explain what is meant by the following terms: (*a*) Radiant energy, (*b*) the ideal black body, (*c*) a gray body, (*d*) the coefficient of emissivity of a body, (*e*) Prevost's Theory of Exchanges, (*f*) Kirchhoff's Law, (*g*) the Stefan-Boltzmann Law, and the modification of it for materials other than "black bodies."

23. (*a*) Calculate the rate at which energy would be emitted from each square foot of a perfect "black body" at temperatures of 0, 100, 500, 1000, 2000, and 3000 F. (*b*) Is this total rate affected by the presence of other bodies? Why?

24. (*a*) If the fuel bed in a locomotive furnace has a temperature of 2400 F, find the net rate of absorption of radiant energy, in B.t.u. per hr., by 10 sq. ft. of boiler heating surface that is parallel to the fuel bed and close enough to it to have an angle factor of 0.5. The emissivity factor is 0.9 and the temperature of the heating surface is 400 F. (*b*) Find the per cent increase in the rate of absorption if the temperature of the fuel bed is increased 10 per cent. (*c*) The same as (*b*) except the temperature of the heating surface is also increased 10 per cent.

Ans. (*a*) 514,200. (*b*) 38.4. (*c*) 38.2.

25. In an experimental determination of the rate of heat transmission through boiler tubes a single tube was supported inside of a fire clay flue with only a small clearance between them. The coefficient of emissivity of the tube was 0.9, and of the flue, 0.95. The outer surface of the boiler tube had an area of 10.55 sq. ft. The flue was maintained at a temperature of 850 F. Find the net rate of heat transmission, in B.t.u. per hr., by radiation from the flue to the tube, if the latter had an outside surface temperature of (*a*) 300 F, (*b*) 400 F, (*c*) 500 F.

Ans. (*a*) 40,750. (*b*) 37,400. (*c*) 32,700.

26. In problem 25, what details of construction of the apparatus would make the results independent of the amount of radial clearance between the tube and the flue? Why?

27. (*a*) What are the gaseous products of combustion that emit a considerable amount of radiant energy? (*b*) Under what conditions does this radiation become important to the engineer? (*c*) Why is radiation from gases difficult to measure? (*d*) What are probably the most important factors affecting the coefficient of emissivity of the solid particles in powdered coal flames?

28. A powdered fuel furnace is cubical in shape and has water-cooled surfaces ($e = 0.95$) at a temperature of 450 F over all of the six faces. The gases in this furnace have a temperature of 2400 F and have 13 per cent CO_2 and 6 per cent H_2O, by volume. The flames are assumed to extend to within 6 in. of each cold surface. (The effect of the solid particles of burning fuel is to be neglected in this problem, but considered later.) Find the rate of absorption of radiant energy, in B.t.u. per hr., from the CO_2 and H_2O if the width of the furnace is (*a*) 10 ft., (*b*) 15 ft., and (*c*) 20 ft. Ans. (*a*) 7,440,000. (*b*) 17,900,000. (*c*) 33,000,000.

29. For each of the furnaces specified in problem 28, the effective flame thickness, to be used in determining the coefficient of emissivity of the suspended particles by means of Eq. (426), may be taken as two-thirds of the actual thickness, and the value of n in this equation may be taken as 90. Determine the rate of heat absorption, in B.t.u. per hr., from the suspended particles by the cooled surfaces for each furnace. Ans. (*a*) 9,480,000. (*b*) 33,600,000. (*c*) 79,800,000.

30. Assume that the suspended solid particles in each of the furnaces in problem 28 will cause a 15 per cent reduction in the absorption of radiant energy from the CO_2 and H_2O as previously calculated. (*a*) Find the total rate, in millions of B.t.u. per hr., of the absorption of radiant energy from the gases and solid particles by the cold surfaces surrounding each furnace. (*b*) Determine what this total rate of absorption is, in B.t.u. per cu. ft. of furnace volume. (*c*) Find the total rate of absorption, in B.t.u. per sq. ft. of furnace heating surface.

Ans. (*a*) 15.8; 48.8; 107.9. (*b*) 15,800; 14,450; 13,480. (*c*) 26,300; 36,200; 44,800.

31. A bare 5-in. horizontal standard pipe (O.D. = 5.563 in.) is carrying steam at a temperature of 600 F, and the outer surface of the pipe line ($e = 0.9$) may also be considered to be at this temperature. (*a*) If the air surrounding this pipe were still and at a temperature of 90 F, compute the 24 hr. loss of heat from 100 linear ft. of this line. (*b*) What fraction of this total loss is due to radiation?

Ans. (*a*) 8,370,000 B.t.u. (*b*) 75.8 per cent.

32. (*a*) Explain what is meant by the term, overall coefficient of heat transfer. (*b*) Explain how it may be calculated. (*c*) Under what conditions may this coefficient, as determined from test data, be used in calculations relating to other equipment?

33. In problem 18 the film coefficient of the hot oil in the heat exchanger was found to be 122 B.t.u. per hr. per sq. ft. per deg. fahr. The cold oil in this exchanger is flowing through the annular space between the 1.5 in. and the 2.5 in. standard steel pipes with a velocity of 4 ft. per sec., and a main body temperature of 80 F. For the cold oil the specific heat is 0.52, the thermal conductivity 1.0, the density

56, and the viscosity 15 centipoises. Considering the thermal conductivity of the pipe to be 320, find the value of the overall coefficient of heat transfer in B.t.u. per hr. per deg. fahr. per sq. ft. of the (*a*) inner surface of the tube, and (*b*) outer surface of the tube. Ans. (*a*) 40.1. (*b*) 34.

34. In problem 19, the film coefficient of heat transfer for the condensing steam may be taken as 2000 B.t.u. per hr. per deg. fahr. per sq. ft. of condensing surface. The outside diameter of the brass tubes is 1.0 inch and their mean temperature is about 100 F. (*a*) Using the film coefficients of the water, at the respective velocities of 2, 4, 6, 8, and 10 ft. per sec., as previously found, calculate the corresponding overall coefficients of heat transfer of this condenser, in B.t.u. per hr. per deg. fahr. per sq. ft., based on the outside area of the tubes. (*b*) Plot the results obtained in (*a*) with water velocities as abscissae. Ans. (*a*) 295.2, 457, 575, 668, 743.

35. In problem 21 (*a*) assume that the tubes in the economizer are cast iron with a thickness of 0.75 in., and that the drop in temperature from the inside of the tube to the main body of the water is 10 F. Find (*a*) the drop in temperature across the tube, (*b*) the water film coefficient of heat transfer, (*c*) the overall coefficient of heat transfer.

Ans. (*a*) 4.2 F. (*b*) 213 B.t.u. per hr. per deg. fahr. per sq. ft. (*c*) 4.52 B.t.u. per hr. per deg. fahr. per sq. ft. of gas surface.

36. (*a*) Make suitable sketches to show the variation of temperature of the two fluids with various amounts of heating surface for the five cases that occur in engineering apparatus designed for heat transmission. (*b*) Give the proper caption to each sketch.

37. (*a*) Explain what is meant by the term, *the true mean temperature difference*, involved in heat transmission. (*b*) Under what conditions does its value equal the arithmetic mean of θ_a and θ_b?

38. Find the arithmetic, and the logarithmic, temperature difference in an air preheater in which the temperatures of the gases at entrance and exit were 470 and 300 F, respectively, while the air entered at 101 F and left at 426 F.

Ans. 121.5, 102.7.

39. Find the percentage error involved by using the arithmetic mean temperature difference in calculating the amount of heat transmitted for the following conditions: (*a*) $\theta_a = 10, 20, 30, 40, 50, 70, 100, 150$ deg., and $\theta_b = 10$. (*b*) $\theta_a = 100, 200, 400, 800, 1000, 1500$ deg., and $\theta_b = 100$.

40. (*a*) Plot the results of problem 39 as ordinates with values of θ_a/θ_b as abscissae. (*b*) At what value of θ_a/θ_b does the error become 5 per cent?

41. A surface condenser receives exhaust steam at a temperature of 115 F. The temperature of the condensing water for this condenser is 60 F at entrance and 105 F at exit. (*a*) Find the mean temperature difference. (*b*) Find the weight of dry saturated steam that would be condensed per hr. per sq. ft. of surface, if $U = 300$. (*c*) Neglecting all heat transfer to external bodies what is the efficiency of the condenser surface? Ans. (*a*) 26.4 F. (*b*) 7.72 lb. (*c*) 81.8 per cent.

42. Determine the rate of flow of the condensing water, in lb. per hr., and the amount of the condenser surface, in sq. ft., required to condense 500,000 pounds of steam per hr. for the conditions as specified, (*a*) in problem 41, (*b*) in (*a*) except $t_2 = 110$ F, (*c*) in (*a*) except $t_2 = 100$ F, and (*d*) in (*a*) except $t_2 = 90$ F.

Ans. (*a*) 11,420,000; 64,900. (*b*) 10,280,000; 82,300. (*c*) 12,850,000; 55,600. (*d*) 17,150,000; 44,900.

43. Plot on cross-section paper the results of problem 42 with θ_b as abscissae, and discuss the main economic factors involved by changing the design to give the various values of t_2.

44. In an ammonia-brine cooler, brine ($c_h = 0.68$) is cooled from a temperature of 40 F to 35 F at the rate of 20,000 pounds per hour. If the evaporation of the ammonia takes place at 30 F determine the surface required, assuming $U = 100$.
Ans. 94.3 sq. ft.

45. In an electric transformer the oil ($c_h = 0.4$) is cooled from a temperature of 160 F to 80 F at the rate of 2500 pounds per hr. by using 5000 pounds of water ($c_c = 1.0$) per hour, the water having an initial temperature of 55 F. If the external heat transfers are negligible and $U = 50$, find the exit temperature of the water, the amount of heating surface required, and its efficiency, for (*a*) parallel flow, and (*b*) counterflow.

Ans. (*a*) 71 F, 40.9 sq. ft., 76.2 per cent. (*b*) 71 F, 31.8 sq. ft., 76.2 per cent.

46. In problem 45, determine the temperatures of both fluids after passing over a surface of 31.8 sq. ft. with parallel flow. Ans. Oil, 85.5 F. Water, 69.9 F.

47. (*a*) Find the amount of counterflow surface required in the oil cooler of problem 45 to give an efficiency of 90 per cent, assuming the same initial temperatures as before. (*b*) Find the exit temperature of the oil. (*c*) Find the exit temperature of the water. Ans. (*a*) 52.6 sq. ft. (*b*) 65.5 F. (*c*) 73.9 F.

48. A counterflow economizer has 45,000 lb. of water per hr. enter at a temperature of 100 F while 72,000 lb. of flue gas enter at a temperature of 950 F. Assume that $c_h = 0.25$, $c_c = 1.0$, $U = 6$, and that the external loss of heat is negligible. (*a*) Find the surfaces required for efficiencies of 20, 40, 60, 80 and 95 per cent, respectively. (*b*) Find the exit temperature of the gas for these efficiencies. (*c*) Find the exit temperature of the water for these efficiencies. (*d*) Plot a curve of efficiency vs. surface.

Ans. (*a*) 699, 1683, 3210, 6119, and 12,590 sq. ft. (*b*) 780, 610, 440, 270, and 142 F. (*c*) 168, 236, 304, 372, and 423 F.

49. The same as problem 48, except the gas temperature is 800 F.

50. Determine the most economical thickness of insulation for an 8-in. pipe carrying steam at a temperature of 750 F under the following conditions: The pipe is in use 8000 hours per year, and the value of the heat saved is 40 cents per million B.t.u. The ambient air is still, and at a temperature of 90 F. The thermal conductivity of the insulation is 0.6, the cost of the insulation is 24 cents per sq. ft. 1 in. thick, and the fixed charges are 15 per cent. Ans. 3.5 in.

CHAPTER XXVII

1. What is meant by the term *fuel*? Name the principal natural and prepared fuels, and state which are first in order of importance in engineering.

2. Give a brief outline of the origin of coal.

3. Give, approximately, the portion of energy supplied from various fuels, and from water, in this country.

4. What are the chief points of value obtainable from the curve in Fig. 506?

5. What determinations are made in a proximate analysis of coal? Define each of the constituents thus determined.

6. What determinations are made in an ultimate analysis of coal? Of what value is such an analysis, and how does its value to the engineer compare with that of the proximate analysis?

7. A proximate analysis of a sample of coal "as received" shows the following results: moisture, 5 per cent; volatile, 30 per cent; fixed carbon, 57 per cent; ash, 8 per cent. Reduce this analysis to a basis of dry coal, and also to a basis of combustible.

8. A proximate analysis of "dry coal" shows the following results: volatile, 30 per cent; fixed carbon, 60 per cent; ash, 10 per cent. Assuming the moisture to have been 5 per cent of the coal as received, compute the various percentages in the coal as received. Also compute the analysis on the basis of combustible.

9. Explain what is meant by the available hydrogen of a fuel having all of its oxygen combined with hydrogen in the form of H_2O.

10. What are the two *basic considerations* involved in the various classifications of coal?

11. Give, in the order of development, the ranks of American coals. Which ranks have the highest heating values? Why?

12. Give briefly the chief physical characteristics of each rank of coal?

13. Discuss Ralston's curve.

14. Discuss Ralston's chart.

15. In Parr's system of coal classification what is meant by "unit coal"?

16. In this country the production of anthracite is, approximately, what portion of that of bituminous coal? How has this ratio varied since 1880?

17. What is the approximate distribution of the use of bituminous coal in this country?

18. (*a*) What fraction of the energy delivered by central stations in the U. S. was derived from fuel in 1919 and in 1929? (*b*) The specific coal consumption in 1919 was what fraction of that in 1929? (*c*) The total energy delivered by the fuel stations in 1929 was what fraction of that in 1919?

19. Name and locate the principal coal fields in the United States, stating what kind of coal is found in each.

20. (*a*) Using the proximate analysis given in Table XXIX for Pocahontas coal (W. Va.), determine the heating value of the combustible of this coal by means of Evans's empirical equation. (*b*) The value obtained in (*a*) is what fraction of that given in the table? (*c*) Calculate by Evans's equation the portion of total carbon in the combustible. (*d*) Determine the percentage error in (*c*) by comparing with the results obtainable from the ultimate analysis.

21. Explain what is meant by the terms, *free ash* and *fixed ash*?

22. What are the disadvantages of using coal with a high percentage of ash?

23. What is clinker? Its formation depends chiefly on what?

24. Classify coals as to the fusibility of the ash.

25. Give the chief physical characteristics of the following coals: (*a*) anthracite, (*b*) semi-anthracite, (*c*) semi-bituminous, (*d*) bituminous, (*e*) sub-bituminous, and (*f*) lignites.

26. (*a*) How is coke obtained and what are its chief uses? (*b*) What are its constituents?

27. How does the heating value of coke compare with that of the coal from which it is made? Why?

28. What other fuels, aside from coke, may be prepared from coal?

29. (*a*) Why is the storage of coal important? (*b*) What are the factors that tend to produce spontaneous combustion of coal in storage?

30. What are the best ways to store coal so that the danger of spontaneous combustion is small?

31. Why is coal of uniform size generally desirable?

32. What are the main items to be considered in the selection of coal?

33. What is the usual basis on which coal is purchased according to specifications?

34. What is hogged fuel? Charcoal?

35. What is petroleum? How was it probably formed? What are the main elements of which it is composed?

36. Explain how the density of an oil may be expressed.

37. How does the heating value of a light oil compare with that of a heavy one (*a*) per unit weight, and (*b*) per unit volume?

38. Name and locate the principal oil fields of the U. S.

39. Explain how crude oil is transported.

40. What are the chief products of an oil refinery?

41. (*a*) What is fuel oil? (*b*) What are its important properties? (*c*) What are its commercial standards?

42. Determine the ratio of the volume of a given weight of oil at a temperature of t_1 deg. to that at 60 F, by the integration of Eq. (502), Table 33.

Ans. $1+A(t_1-60)+B(t_1-60)^2$.

43. A firm ordered fuel oil of 25 A.P.I. degrees at 60 F, the price agreed upon being $1.50 per bbl. The oil was delivered into a calibrated tank that showed 10,500 bbl. when measured at 180 F. How much should be paid for the oil delivered? Ans. $14,988.

44. A crude oil of 20 A.P.I. degrees at 60 F has a heating value of 19,020 B.t.u. per lb. (*a*) Find the heating value in B.t.u. per gallon measured at 60 F. (*b*) The same as (*a*) except the temperature is 90 F. (*c*) What is the gravity of this oil in Baumé degrees at 60 F? (*d*) Find the per cent error involved by using Eq. (501) to determine the heating value.

Ans. (*a*) 148,000. (*b*) 146,200. (*c*) 19.89. (*d*) 0.16.

45. (*a*) Determine how much heat must be supplied to a tank car to heat 10,000 gallons of fuel oil, measured at 60 F, from an initial temperature of 20 F to a final temperature of 180 F. It is assumed that 10 per cent of the heat delivered to the heating coils is lost to the atmosphere. The specific gravity of the oil is 0.9 at 60/60 F. (*b*) The heat supplied to warm the car load of oil as in (*a*) is what fraction of that obtainable from its combustion? (*c*) What is the mean specific heat of this oil, in B.t.u. per gal. and per lb., for the range of temperature involved in this problem?

Ans. (*a*) 6,094,000 B.t.u. (*b*) 0.42 per cent. (*c*) 3.43, 0.457.

46. A large steamship consumes 1.3 tons of fuel oil per mile when traveling at an average speed of 26 knots. The oil used has a heating value of 19,500 B.t.u. per lb., and a specific gravity of 0.865. (*a*) How much space is occupied by the oil used by such a vessel during a 2000 mile trip? (*b*) If a similar vessel used bituminous coal just as efficiently as the oil is used in (*a*), what space would be required for the coal consumed during the same trip? The coal has a heating value of 14,500 B.t.u. per lb., and weighs 50 lb. per cu. ft. (*c*) If the oil costs \$1.50 per bbl., what would be the price of the coal per ton (2000 lb.) to equal that of the oil on the energy basis alone? (*d*) When the cost of the two fuels per unit of energy is the same what other items enter into the question of the choice of these two fuels for a steamship in any particular case?

Ans. (*a*) 96,500 cu. ft. (*b*) 139,900 cu. ft. (*c*) \$7.37.

47. Define the following terms and discuss the significance of each in the oil industry: (*a*) flash point, (*b*) pour point, (*c*) viscosity, and (*d*) distillation test.

48. (*a*) Compare the general properties and uses of kerosene and gasoline. (*b*) Which has the higher heating value per unit weight and also per unit volume?

49. A certain gasoline gives a hydrometer reading of 66.5° Bé at 60 F. (*a*) Find the specific gravity and the pounds per gallon of this gasoline at 60 F. (*b*) Find the H.H.V. of this gasoline by Eq. (503). (*c*) By means of the results of (*a*) and (*b*) find the H.H.V. in B.t.u. per gal. (*d*) The result obtained in (*c*) is what per cent of that found by using Eq. (504)? (*e*) The result obtained by using equation (503) is what per cent of that found by using Eq. (515)?

Ans. (*a*) 0.712, 5.94. (*b*) 20,400 B.t.u. per lb. (*c*) 121,200. (*d*) 100 per cent. (*e*) 99.2 per cent.

50. What are some of the fuels that may be blended with gasoline to improve its detonating characteristics?

51. Discuss the terms, *cracking* and *hydrogenation processes*, as applied in the petroleum industry.

52. Discuss *coal carbonization*, and *coal liquefaction* by *hydrogenation*.

53. Explain why alcohol is not extensively used as a motor fuel in this country at the present time.

54. (*a*) Where are the principal natural gas fields in the United States? (*b*) What are the chief constituents of natural gas?

55. Explain how the following gases are made: producer gas, coke-oven gas, and blue water gas.

56. Compare the heating values of natural gas, blast furnace gas, producer gas, coke-oven gas, and water gas.

57. (*a*) What is the cost of 1,000,000 B.t.u. of energy available in the furnace of a residential heating plant in your own community with the various fuels there available at present prices? (*b*) Discuss how the maintenance of clean heating surfaces affects the utilization of the available energy from each of these fuels. (*c*) What other factors should also be considered when selecting a fuel for this use? (*d*) If you do not have the necessary data for your own community answer (*a*), (*b*), and (*c*) for the following cases:

Fuel	H. H. V.	Price
Natural gas	1,000 B.t.u. per cu. ft.	\$0.50 per 1000 cu. ft.
City gas	540 B.t.u. per cu. ft.	1.00 per 1000 cu. ft.
Fuel oil	148,000 B.t.u. per gal.	0.07 per gal.
Anthracite coal	12,800 B.t.u. per lb.	10.00 per ton
Bituminous coal	13,500 B.t.u. per lb.	5.00 per ton.

58. Describe the present condition of the anthracite and bituminous coal industries.

59. Describe the present condition of the petroleum industry.

60. Discuss some of the probable future developments regarding fuels.

CHAPTER XXVIII

1. What is meant by *endothermic reactions*? *Exothermic ones*?

2. The *heat of reaction* is also designated by what other terms when dealing with combustion?

3. In engineering work what is the purpose of combustion? What are the usual combustibles?

4. Define the following terms: slow combustion, rapid combustion, ignition temperature, and spontaneous ignition.

5. What factors determine the rapidity of ignition, and the rate of combustion of (*a*) solid and liquid fuels, and (*b*) gaseous fuels?

6. What are the four conditions to be fulfilled to insure complete combustion of a fuel?

7. What is meant by the term *exit gases* when applied to combustion?

8. Explain what is meant by the term *atomic weight*. Give its numerical value for each of the five most important elements involved in combustion.

9. Discuss the terms, *molecular weight* and *formula weight*, as applied to the different phases of a substance.

10. (*a*) Determine the number of pounds represented by 1 *lb.f.w.* of C_8H_{18}? (*b*) How many grams are represented by 1 *g.f.w.* of $C_{12}H_{26}$?

11. (*a*) What three important relations are expressed by a chemical equation applied to the combustion of a gas? (*b*) Discuss these relations when the equation is applied to the combustion of solids or liquids at atmospheric pressure.

12. Discuss the chief effects of the nitrogen in the air used for combustion.

13. Define the terms, *high heating value* and *low heating value*.

14. What are the units used to express the heating values of different kinds of fuels?

15. The *standard condition* of a gas is determined by what pressure and temperature?

16. Explain how the high or low heating value of fuels may be experimentally found.

17. From the International Critical Tables the high heating values of C_3H_8 and $C_{10}H_8$, at constant pressure and an initial temperature of 18 C, are given as 526.3 and 1231.8 kg. cal. per gram mole, respectively. Find the corresponding values of (*a*) H.H.V. in B.t.u. per lb. f.w., (*b*) H.H.V. in B.t.u. per lb., (*c*) L.H.V. in B.t.u. per lb. Ans. (*a*) 947,340, 2,217,240. (*b*) 21,530, 17,320. (*c*) 19,800, 16,730.

18. Calculate the difference between the gross and net heating values at constant pressure of C_8H_{18} at an initial temperature of (*a*) 60 F, (*b*) 70 F. Compare the results of (*a*) and (*b*) with the one obtained from Table XXXVII, and discuss the general significance of this comparison.

Ans. (*a*) 1505 B.t.u. per lb. (*b*) 1497 B.t.u. per lb.

19. Write the chemical equation for the complete combustion with pure oxygen of the following fuels: (*a*) Methane gas (CH_4), (*b*) Acetylene gas (C_2H_2), (*c*) Liquid benzene (C_6H_6), (*d*) Liquid octane (C_8H_{18}). Determine the relative weights and volumes involved in each equation, assuming that the volume of each of the liquid fuels is negligible relative to that of the oxygen, and that none of the vapor is condensed.

20. (*a*) Find the actual ratio of the volume of the fuel in (*c*) and (*d*) of the previous problem to that of the oxygen, if both are measured at a pressure of 1 atmosphere and a temperature of 60 F, the specific volume of the C_6H_6 being 0.0182 cu. ft. per lb. and that of the octane being 0.0217 cu. ft. per lb. (*b*) Would this ratio be materially changed if the fuel and oxygen were both at a pressure of 50 atmospheres and the same temperature as before? Ans. (*a*) 1/2000, 1/1910.

21. Considering air to be a mixture of 23.2 per cent O_2 and 76.8 per cent N_2, by weight, find the weight of air required to burn 1 lb. of each of the fuels specified in problem 19. Write the reaction equation for 1 lb. of each fuel burned with just sufficient air to supply the necessary oxygen.

Ans. (*a*) 17.25 lb. (*b*) 13.28 lb. (*c*) 13.28 lb. (*d*) 15.14 lb.

22. (*a*) If 100 lb. of carbon are burned in a furnace with 240 lb. of oxygen determine the weight of CO_2 and CO formed. (*b*) The heat developed by these reactions is what per cent of that produced with sufficient oxygen to burn all of the carbon to CO_2? (*c*) Discuss the general significance of the results of (*a*) and (*b*) as applied to the combustion of carbon. Ans. (*a*) 293.3, 46.7. (*b*) 85.6 per cent.

23. A certain coal has the following composition: C = 0.85, H_2 = 0.04, N_2 = 0.01, O_2 = 0.02, S = 0.01, and ash = 0.07. (*a*) Write the chemical equation for the combustion of 1 lb. of this coal with pure oxygen. (*b*) If this coal is burned at the rate of 2 tons per hr. with 50 per cent excess air what is the weight of dry air required per hr.? (*c*) What is the volume of the air, in cu. ft. per min., that must be handled by a fan that supplies the amount of air as in (*b*) if measured at a temperature of 70 F and a pressure of 14.7 lb. per sq. in. abs. (*d*) Determine the percentage composition by weight of the products of combustion when using 50 per cent excess air. (*e*) How much SO_2 would pass out the smoke stack each month from the furnace burning this coal at the rate specified in (*b*)? (*f*) How much ash would pass out the stack each month if the rate of combustion specified in (*b*) is maintained and if 50 per cent of the ash goes out the stack?

Ans. (*b*) 66,700 lb. (*c*) 14,830. (*e*) 57,600 lb. (*f*) 100,800 lb.

24. What are the products of combustion likely to be when a fuel composed of carbon, hydrogen, and sulphur is burned with a deficiency of air? Why?

25. Describe the different zones that exist in the furnace used to burn coal.

26. (*a*) What are the chief reactions taking place in manufacturing producer gas? (*b*) What is meant by the term *equilibrium ratio* between C, CO and CO_2? What factors affect this ratio?

27. (*a*) Rewrite equation (*a*), page 374, for 1 lb. of carbon, using the atomic weights given in Table XXXVII. (*b*) Using this table to obtain the heating values of hydrogen and carbon when burned to H_2O and CO, respectively, prove that your result from (*a*) is correct.

28. Discuss the combustion of hydrocarbons as to (*a*) the ultimate product formed, (*b*) the reactions involved, and (*c*) the heating value.

29. Prove that the general expression given by Eq. (529) is closely correct for the weight of oxygen required to burn 1 lb. of C_nH_m.

30. (*a*) Determine the ratio of the heating value of H_2 to that of CO, when measured on the volumetric basis. (*b*) If each of these gases is mixed with just sufficient air to complete combustion, find the ratio of the resulting heating values per unit volume of the mixtures.

31. Discuss the chief difficulties resulting from the burning of sulphur.

32. Explain what is meant by *chemical equilibrium* and by *dissociation.*

33. Discuss the significance of the curves shown in Fig. 525.

34. (*a*) Increasing the pressure of CO_2 from 0.1 to 1 atmosphere has what effect on its dissociation if the temperature is kept constant at 3500 F. (*b*) The same as (*a*) except the gas is H_2O.

35. Using the results given in Table XXXIX (p. 382), determine how the amount of dissociation of CO_2 and H_2O in the ideal Otto engine is affected by (*a*) the mixture, (*b*) the compression ratio, and (*c*) the period in the cycle at which the dissociation is calculated.

36. The same as the previous problem, except the engine is a Diesel.

37. Is the amount of dissociation in an actual Otto or Diesel engine greater or less than that calculated for the corresponding ideal one? Why?

38. In calculating the maximum temperature attained during combustion in a furnace or engine what heating value should be used? Why?

39. What are the items that affect the maximum temperature attained during combustion in (*a*) the furnace of a steam-generating unit, (*b*) the cylinder of a Diesel engine?

40. Discuss the important points shown by the data given in Table XL (p. 386).

41. Discuss the effect of the initial pressure of a gaseous mixture upon the maximum temperature attained during combustion.

42. What are the factors that have an important effect on the ignition of carbon monoxide?

43. (*a*) Define the terms, *higher* and *lower limits of inflammability* of a gaseous mixture. (*b*) How are these limits affected by using oxygen instead of air to mix with the fuel?

44. A gaseous fuel mixture is composed of 20 per cent CO, 10 per cent H_2 and 70 per cent CH_4, by volume. Determine the maximum and minimum volumes of air that might be mixed with 10 cu. ft. of this fuel mixture and still have an inflammable one. All volumes are to be measured at atmospheric pressure and at a temperature of 70 F. The limits of inflammability, as given in Table XLI (p. 392), are to be used. Ans. 162.5 cu. ft., 44.1 cu. ft.

45. What other factors, in addition to the composition of the mixture, affect the inflammability of gases and vapors?

46. How is the range of inflammability of gases and vapors affected by (*a*) reducing the pressure far below atmospheric, and (*b*) increasing the pressure far above atmospheric?

47. Discuss the effect of temperature on the range of inflammability of a mixture.

48. Explain why the direction of flame propagation affects the range of inflammability, as indicated by the curves in Fig. 530.

49. Explain the significance of the terms, *normal burning* and *detonation wave.*

50. (*a*) Explain what is meant by velocity of flame propagation. (*b*) This velocity is of what order of magnitude with *normal burning* and with *normal detonation*?

51. Trace briefly the zone of reaction as it probably develops within the combustion chamber of an Otto engine (*a*) without detonation, and (*b*) with detonation.

52. What are some of the theories advanced to explain the manner in which a detonation wave spreads so rapidly?

53. Discuss the effect on detonation in an Otto engine of (*a*) the compression ratio, (*b*) the time of ignition, (*c*) the size of the cylinder, (*d*) the shape of the combustion chamber, (*e*) the fuel used, and (*f*) throttling.

54. (*a*) Using the data as given by the curves in Fig. 532, determine what spark advance should be used with a compression ratio of 6 to 1 to give the same power, without much detonation, as the engine having a compression ratio of 4.2 to 1 and a spark advance of 47 degrees. (*b*) What other spark advance could be used by the latter engine without decreasing its power? (*c*) Would this produce more or less detonation?

55. (*a*) Explain what is meant by *preignition* in an engine. (*b*) What may cause it in an Otto? (*c*) In a Diesel?

CHAPTER XXIX

1. What information of value to the engineer may be obtained from a quantitative analysis of the exit gases from a combustion chamber?

2. What does the engineer generally mean by the terms, *products of combustion*, *exit gases*, *flue gases*, and *exhaust gases*?

3. (*a*) In making an analysis of exit gases, describe the method of collecting the sample and explain the importance of this. (*b*) What determinations are generally made? (*c*) Describe the usual method of procedure.

4. Explain how the results from the Orsat analysis are the same as though a *dry gas* had been analyzed.

5. Explain why the analysis of the exhaust gases from an automotive engine can not generally be made satisfactorily by means of the usual Orsat apparatus.

6. The analysis of a certain mixture of gases shows the following percentages by volume: $H_2 = 10$, $CO = 24$, $CO_2 = 11$, and $N_2 = 55$. (*a*) Find the composition of this mixture by weight percentages. (*b*) Find the weight of 1000 cu. ft. of this gas when at a temperature of 60 F and a pressure of 14.7 lb. per sq. in. abs. (*c*) Find the higher heating value of this gas, in B.t.u. per lb. and in B.t.u. per cu. ft., at the pressure and temperature specified in (*b*). (*d*) What percentage of the heating value of this gas is due to each constituent? (*e*) Find the weight and volume of dry air (at 14.7 and 60 F) required to burn completely the gas specified in (*b*).

Ans. (*b*) 71.55 lb. (*c*) 1529, 109. (*d*) 29.4, 70.6. (*e*) 61.8 lb., 813 cu. ft.

7. Assume the 1000 cu. ft. of the gaseous mixture given in the previous problem to be burned with just sufficient dry air to support combustion. (The low pressure steam in the products may be considered to follow the law, $PV = wRT$, with $R = 85.7$.) (*a*) Determine the percentage composition, by weight and by volume, of the products of combustion when at a temperature of 540 F and a pressure of 1 atmosphere. (*b*) Find the partial pressure, in lb. per sq. in. abs., of each constituent in the products mixture in the condition given by (*a*). (*c*) Explain what happens to each partial pressure and each percentage by volume, when the mixture is cooled at constant pressure. (*d*) Find the temperature at which condensation would begin if the products mixture were kept at a pressure of 1 atmosphere.

Ans. (*b*) $H_2O = 0.89$, $CO_2 = 3.11$, $N_2 = 10.7$. (*d*) 98 F.

8. (*a*) If the products mixture of the previous problem were kept at a pressure of 1 atmosphere and the temperature reduced to 60 F, find the weight of steam that would be condensed from the products mixture for each 1000 cu. ft. of the fuel gas (at 14.7 and 60 F). (*b*) The value determined for (*a*) is what fraction of the H_2O formed by the combustion of the hydrogen? (*c*) Trace on a temperature-entropy diagram the various states of a unit weight of vapor in the mixture, assuming all condensation to be drained off as fast as formed.

Ans. (*a*) 3.45 lb. (*b*) 72.6 per cent.

9. The products of combustion of bituminous coal in a certain case had the following percentages *by weight*: $CO_2 = 17.5$, $O_2 = 7$, $N_2 = 72$, and steam = 3.5. For the low-pressure steam, $R = 85.7$, $M' = 0.463$, $B = -1.53(10)^{-5}$ and $C = 2.35(10)^{-8}$. (*a*) Using additional values, as given in Table *E* of the Appendix find the values of the constants, M'_m, B_m, and C_m for the products mixture. (*b*) Assuming these products to pass over the heating surfaces of a boiler at the rate of 10,000 lb. per min. and to be thereby cooled from 2440 F to 540 F, find the rate of heat absorption from the products mixture. (*c*) Find the mean value of the specific heat at constant pressure of this mixture for the range of temperature given in (*b*). (*d*) Determine the per cent error that would have resulted in (*b*) if 0.24 had been taken as the mean value of c_p.

Ans. (*a*) 0.238, $1.496(10)^{-5}$, $1.789(10)^{-9}$. (*b*) 5,215,000 B.t.u. per min. (*c*) 0.2745. (*d*) 12.6 per cent.

10. If the analysis of the products of combustion in the previous problem could have been made without error by means of the usual Orsat apparatus, what would have been the results of such an analysis?

Ans. $CO_2 = 12.4$ per cent, $O_2 = 6.8$ per cent, $N_2 = 80.8$ per cent.

11. (*a*) If the products of combustion in problem 9 had been discharged from the last heating surface of a boiler at 540 F, find the rate at which these products would still give up heat in passing over additional surfaces that reduce their temperature to 140 F. (*b*) Find the per cent error that would have resulted in (*a*) if the mean value of c_p had been taken as 0.24. (*c*) Find the mean value of c_p of the *dry gases* in (*a*). (*d*) Discuss the general significance of the results of (*b*) and (*c*); also those of (*d*) in problem 9.

Ans. (*a*) 1,004,500 B.t.u per min. (*b*) 4.4 per cent. (*c*) 0.2435.

12. An Orsat apparatus gave the following analysis of flue gas: $CO_2 = 0.13$, $CO = 0.005$, $O_2 = 0.07$, and $N_2 = 0.795$. The analysis of the coal used showed that it contained 75 per cent carbon and 6 per cent ash. The test of the ash-pit refuse showed that it contained 20 per cent unconsumed carbon. For each pound of fuel fired, find (*a*) the weight of carbon actually burned, and (*b*) the weight of dry flue gas.

Ans. (*a*) 0.735 lb. (*b*) 13.8 lb.

13. (*a*) The same as problem 12, except the ash-pit refuse contains 50 per cent unconsumed carbon. (*b*) Discuss the significance of accuracy in determining the unburned carbon in the refuse.

Ans. (*a*) 0.69 lb., 12.9 lb.

14. Show that Eq. (549) is true.

15. Prove that Eq. (550) holds for most coals.

16. Find the weight of air supplied per lb. of coal fired for the data given in problem 12, assuming that the coal contained (*a*) no nitrogen, and (*b*) 1.6 per cent nitrogen. Ans. (*a*) 13.16 lb. (*b*) 13.14 lb.

17. If a flue gas analysis shows both O_2 and CO, what conclusions may be reached?

18. (*a*) Why is *excess air* usually supplied to a coal-burning furnace? (*b*) Define *excess coefficient* and state what other terms are used for it.

19. If pure carbon is completely burned with air, calculate from the chemical equation the percentage, by volume, of the CO_2, O_2, and N_2 in the exit gases when the dilution coefficient is (*a*) 1.00, and (*b*) 1.50. Check the results by means of Fig. 533 and Eq. (556). Ans. (*a*) 20.9, 0, 79.1. (*b*) 13.93, 6.96, 79.1.

20. Does a very high value of CO_2 in the flue gas always indicate satisfactory combustion? Why?

21. Find the dilution coefficient when 15 lb. of air are used per lb. of coal having the following composition: $C = 0.83$, $H_2 = 0.05$, $O_2 = 0.04$, $S = 0.01$, and ash $= 0.07$. Ans. 1.35.

22. Discuss fully the results shown by the curves in Fig. 534 (p. 418).

23. Explain fully by the aid of Fig. 535 how the *optimum values* of CO_2 and of the dilution coefficient may be obtained for a given fuel and operating conditions.

24. Explain why the value of O_2, or N_2, in the exit gas is, in general, a more accurate measure of the excess air than the CO_2.

25. An Orsat analysis of the flue gas from a coal-burning furnace gave the following results: $CO_2 = 0.12$, $CO = 0.01$, $O_2 = 0.07$, and $N_2 = 0.80$. (*a*) Find the percentage of excess air used. (*b*) Determine the weight of air used per pound of carbon burned. (*c*) Find the weight of dry flue gas per pound of carbon burned. (*d*) Calculate the ratio, by weight, of nitrogen to total oxygen (free or combined) in the dry flue gas. (*e*) Check the result of (*a*) by Fig. 536.

Ans. (*a*) 44.4. (*b*) 18.73 lb. (*c*) 19.4 lb. (*d*) 3.59.

26. The exhaust from an internal combustion engine involves what three energy losses?

27. The water vapor in exit gases may come from what sources?

28. From the values given in the Keenan Steam Table (see Appendix), calculate the mean value of c_p for superheated steam at a pressure of 1 lb. per sq. in. abs. and a range of temperature from saturation to (*a*) 500 F, and (*b*) 600 F.

Ans. (*a*) 0.458. (*b*) 0.461.

29. Calculate the weight of water vapor accompanying 1000 lb. of dry air in the atmosphere having a temperature of 80 F. and a pressure of 14.7 lb. per sq. in., if the relative humidity is (*a*) 90 per cent, (*b*) 70 per cent, and (*c*) 50 per cent.

Ans. (*a*) 19.96 lb. (*b*) 15.4 lb. (*c*) 10.93 lb.

30. A certain coal, having a high heating value of 13,230 B.t.u. per lb. as fired, has the following composition: $C = 0.76$, $H_2 = 0.05$, $O_2 = 0.06$, $N_2 = 0.01$, ash $= 0.08$, and moisture $= 0.04$. Assume that this fuel is completely burned with 50 per cent excess air, both supplied at atmospheric temperature of 80 F. The relative humidity of the atmosphere is 90 per cent and the exit gas leaves the last heating surface at a temperature of 500 F. Find what percentage of the high heating value of the fuel is lost due to (*a*) the dry exit gas, (*b*) the moisture in the fuel, (*c*) the vapor from the hydrogen burned, (*d*) the vapor from the atmosphere, and (*e*) the total products to the stack.

Ans. (*a*) 11.98. (*b*) 1.01. (*c*) 3.58. (*d*) 0.445. (*e*) 17.

31. The same as problem 30, except the relative humidity is only 50 per cent.

Ans. (*a*) 11.98. (*b*) 1.01. (*c*) 3.58. (*d*) 0.24. (*e*) 16.81.

32. If the exit gases of problem 31 could be cooled down to 80 F by some form of useful heat-absorbing surfaces, find (*a*) the vapor pressure in the exit gases at 500 F if the total pressure is 14.7 lb. per sq. in. abs., (*b*) the temperature at which condensation of this vapor would begin, (*c*) the total weight of vapor per lb. of fuel fired before any condensation takes place, (*d*) the weight of vapor condensed, and (*e*) the heat absorbed by all of the heating surfaces, in per cent of the H.H.V. Discuss part (*e*).

Ans. (*a*) 0.963 lb. per sq. in. abs. (*b*) 100.5° F. (*c*) 0.657 lb. (*d*) 0.32 lb. per lb. of fuel fired. (*e*) 98.7.

33. (*a*) Explain what is meant by the dew point of flue gases containing H_2O and SO_3. (*b*) Calculate the amount of sulphuric acid (H_2SO_4) that could be made from 1 ton of sulphur. (*c*) What conditions are most favorable to the corrosion of metal surfaces exposed to flue gases? Ans. $3\frac{1}{16}$ tons.

34. (*a*) Explain how exit gases may cause damage. (*b*) Explain how sulphur products may be removed from exit gases.

35. (*a*) What is smoke? (*b*) How may its density be measured? (*c*) How may solid materials be removed from exit gases?

CHAPTER XXX

1. What is the source of energy used to generate nearly all of the steam used in power plants? Why?

2. (*a*) The usual type of steam-generating unit performs what two basic functions? (*b*) What are the main elements that make up such a unit?

3. Make a sketch of a horizontal return tubular boiler and furnace.

4. Make a sketch of a water tube boiler and furnace.

5. What is the distinction between integral and independent types of economizers?

6. What is the primary purpose of (*a*) air preheaters, (*b*) forced-draft fans, and (*c*) induced-draft fans?

7. Name the purposes for which the various auxiliaries of a steam-generating unit are used.

8. Is the feed pump an auxiliary of a steam-generating unit? Why?

9. In measuring the amount of heating surface in any part of a steam-generating unit, what one basic rule is applied?

10. What heating surfaces are included in the (*a*) boiler, (*b*) superheater, (*c*) economizer, (*d*) reheater, and (*e*) air heater?

11. For a steam-generating unit, explain what is the most logical method of expressing (*a*) its size, (*b*) its capacity, and (*c*) its load.

12. (*a*) If the manufacturer of a small boiler should list it as 300 h.p., what would he mean? (*b*) Why is this method not followed by the manufacturer of large units?

13. (*a*) What was meant by the old term "normal rating" of a boiler? (*b*) Why is this term not used by the best engineers in connection with large steam-generating units today?

14. (*a*) Explain what is the meaning of the efficiency of a steam-generating unit. (*b*) Distinguish between the gross and net efficiencies of such a unit.

15. Derive the equation for the net efficiency of a steam-generating unit having no reheater.

16. Explain how the equation derived in problem 15 should be modified if the unit contains (*a*) a *gas reheater*, (*b*) a *steam reheater* of the type shown in Fig. 539, and (*c*) a *steam reheater* located at the turbine and receiving its heat from steam taken from the main line leading to the turbine.

17. The feedwater enters a steam-generating unit with a heat content of 400 B.t.u. per lb. The steam leaves the unit with a pressure of 1400 lb. per sq. in. abs., a temperature of 750 F, and at the rate of 498,000 lb. per hr. The unit consumes powdered coal, having a H.H.V. of 11,800 B.t.u. per lb., at the rate of 44,800 lb. per hr. The auxiliaries of the unit are all electrically driven and require 55 kw-hr. per ton of coal fired. The net fuel rate of the station is 1.18 lb. per kw-hr., and 5 per cent of the gross generator output is used to operate all of the station auxiliaries. (*a*) Find the gross and net efficiencies of the steam-generating unit for the above conditions. (*b*) The same as (*a*), except the energy consumption of the auxiliaries of the steam-generating unit is assumed to be 20 per cent less. (*c*) The same as (*a*), except the steam generated per pound of fuel used is assumed to be 3 per cent greater. (*d*) If you were in charge of such a unit, what investigations would you make to determine the feasibility of attempting to attain the result specified in (*c*)?

Ans. (*a*) 88, 85.3 per cent. (*b*) 88, 85.8 per cent. (*c*) 90.6, 87.8 per cent.

18. Explain how to calculate closely the heat content of the feedwater entering a steam-generating unit at any given pressure, and at a temperature less than 300 F. At such temperatures the volume of the water may be considered to remain constant during compression.

19. (*a*) Determine the heat content of 1 lb. of feed water having a pressure of 1500 lb. per sq. in. abs. and a temperature of 281 F. (*b*) The same as (*a*), except the pressure is 2000.

20. (*a*) Name the losses that occur in a steam-generating unit. (*b*) Discuss the relative magnitude of these losses.

21. (*a*) Explain what is meant by the *energy balance* of a steam-generating unit. (*b*) This balance requires what test data? (*c*) Make a diagram to show all of the means by which energy may enter or leave such a unit.

22. Draw, approximately to scale, a diagram to show the magnitude of the various streams of energy leaving the steam-generating unit, for which the test data are given in Table XLVII (p. 454). (*a*) Use Test No. 6. (*b*) Use Test No. 9.

23. Explain how to derive the equations for items (1), (2), and (3), in Table XLVI (p. 448).

24. Discuss the results shown by the curves in Fig. 542.

25. Using the curves shown in Fig. 543, explain (*a*) why the air preheater increases the efficiency of the unit, and (*b*) why the carbon loss increases so rapidly at the very high rates of evaporation.

26. Discuss the results shown by the curves in Fig. 544.

27. (*a*) In the steam-generating unit No. 22, Calumet Station, the heating surface of the air preheater is what portion of the total? (*b*) What are the advantages and disadvantages of such a large air heater? (*c*) Compare the amount of heating surface in the economizer of this unit with that of the air heater, and then from the test data compare the relative rates of heat transmission. Explain.

28. (*a*) Using the data from Tests 1 to 9 inclusive, Table XLVII, draw curves for the gross and net efficiencies, with the rate of heat liberation per unit volume of furnace as abscissa. (*b*) Explain, by the aid of these test results, the chief causes of the variation in the net efficiency shown in (*a*).

29. Discuss the results shown by the curves in Fig. 546.

30. (*a*) What are the factors that affect the net efficiency of a steam-generating unit? (*b*) What other factors must also be considered in selecting a steam-generating unit for any specific case? Why?

CHAPTER XXXI

1. Discuss briefly the basic requirements to be fulfilled to secure the complete combustion of coal when burned on a grate.

2. (*a*) If a 6-in. bed of coal is burned on a grate, indicate by a sketch about how the percentages of CO_2, CO, and O_2 vary with the distance above the grate. (*b*) Give the reasons for the results shown in (*a*).

3. The same as problem 2, except the fuel bed is 12 in. thick.

4. What part of the furnace has the highest temperature when coal is burned on a grate? Why?

5. When coal is burned on a grate, what controls (*a*) the *rate of burning*, and (*b*) the *completeness of combustion*?

6. What conditions must prevail to attain a very high temperature within the fuel bed?

7. Determine the percentage error involved in using the empirical Eq. (572) to find the weight of air just sufficient to burn (*a*) Pocahontas coal (see p. 324), (*b*) New River coal, (*c*) Illinois coal, and (*d*) Colorado bituminous.

8. When burning coal on a grate, (*a*) why is excess air desirable, (*b*) on what does the desired amount of excess air depend, (*c*) how may the most satisfactory amount of air be found, and (*d*) what percentages of excess air are commonly used?

9. Show by means of curves how the optimum percentage of excess air may be calculated for a boiler furnace.

10. (*a*) Name the items affected by furnace temperature. (*b*) How is the furnace temperature controlled?

11. Explain how the fuel entering a furnace may be subjected to the proper temperature.

12. What are the difficulties encountered if the furnace temperature becomes too high?

13. Increasing the furnace temperature has what effect on the absorption of heat by the (*a*) furnace walls, (*b*) boiler tubes, (*c*) superheater, and (*d*) economizer?

14. The amount and proportions of the combustion space needed within a furnace depend mainly on what three items? Why?

15. What two conditions should be fulfilled to obtain the proper size and arrangement of the combustion space?

16. What are some of the special means used to mix the secondary air and gases?

17. Do the physical characteristics of a coal have to be considered when selecting a stoker, or when deciding what method of hand firing to use? Why?

18. The curves given in Fig. 549 for the Pittsburgh run-of-mine coal, fired at the rate of 50 lb. per hr. per sq. ft. of grate, show that the flame should travel about how far, if the loss due to incomplete combustion of the gases is not to exceed 4 per cent when the excess air is (*a*) 15 per cent, (*b*) 25 per cent, and (*c*) 50 per cent.

19. The same as problem 18, except the rate of firing is half as much.

20. What are the general conclusions to be drawn from the results of problems 18 and 19?

21. Using the chart given by Fig. 550 for Pocahontas coal burned at the rate of 50 lb. per hr. per sq. ft. of grate, with 50 per cent excess air and a loss of 5 per cent due to the incomplete combustion of the gases, determine what should be the approximate values of (*a*) the combustion space per sq. ft. of grate and (*b*) the length of flame travel.

22. The same as problem 21, except the loss due to incomplete combustion is to be only 1 per cent.

23. The same as problem 21, except the excess air is only 10 per cent, and the loss due to incomplete combustion is 1 per cent.

24. The same as problem 21, except the coal is to be Illinois.

25. Compare the results found in problems 21 and 24 with the data given in Table 48.

26. Determine, from the curves in Fig. 551, the optimum percentage of the excess air and the corresponding percentage of unavailable energy for a boiler, assuming Illinois coal to be burned at the rate of 50 lb. per hr. per sq. ft. of grate with a ratio of combustion space to grate area of (*a*) 7, (*b*) 5, and (*c*) 3.

27. The same as problem 26, except the fuel is Pocahontas coal.

28. Using the curves in Fig. 552, determine what the percentage of excess air should be to burn Pittsburgh coal at the rate of 50 lb. per hr. per sq. ft. of grate with minimum losses, if the ratio of the combustion space to the grate area is (*a*) 7, (*b*) 5, and (*c*) 3.

29. Determine the percentage of CO_2 corresponding to each case in problem 28.

30. What are the important factors to be considered in selecting combustion equipment?

31. (*a*) Determine the saving in the annual cost of fuel if a large central station can reduce its specific coal consumption from 1.04 to 0.988 lb. per kw-hr. when operating with an average yearly load of 200,000 kw. Assume that the coal costs $5.00 per ton. (*b*) Under what conditions might the result obtained in (*a*) be a net saving to the company? Why? (*c*) Under what conditions might the net saving be zero? Why?

32. What 7 conditions should fulfilled to obtain the best results from the operation of a furnace?

33. (*a*) When burning coal on hand-fired grates how may the fuel bed be maintained in the best condition? (*b*) Name and describe the three methods of hand-firing.

34. Discuss the following points involved in the proper operation of a hand-fired furnace: (*a*) apportioning the air used, (*b*) regulating the draft to vary the rate of combustion as needed, (*c*) determining the intervals between firing, (*d*) cleaning the fire, (*e*) avoiding serious clinker trouble, and (*f*) determining the best thickness of the fuel bed.

35. Make a sketch of a roofless combustion chamber and explain why it is suitable for certain fuels.

36. The same as problem 35, except a Dutch oven is to be used.

37. Explain by the aid of a sketch how baffles and arches may be used to form a modification of the Dutch oven.

38. Explain how the Hawley-down-draft furnace operates.

39. (*a*) Explain how to determine the amount of grate area that should be installed for hand firing in any specific case. (*b*) How much fixed carbon is generally burned per sq. ft. of grate with natural draft?

40. Discuss the results shown by the curves in Fig. 561.

41. What factors should be considered in determining the width and length of hand-fired grates?

42. Describe the desirable characteristics of good grate bars.

43. Under what conditions is hand firing justifiable? Why?

44. Give the eight principal advantages obtainable from mechanical stokers as compared with hand firing.

45. Explain what is meant by (*a*) overfeed stokers and (*b*) underfeed stokers.

46. What are the various types of (*a*) overfeed stokers and (*b*) underfeed stokers?

47. (*a*) Make a sketch to show the essentials of an inclined front-feed stoker. (*b*) Explain how this stoker operates. (*c*) Describe the double inclined side-feed stoker.

48. (*a*) What are the principal advantages of the inclined side-feed stoker? (*b*) What kinds of fuels are commonly used in it? (*c*) What are its disadvantages?

49. (*a*) Make a sketch of a chain-grate stoker and describe its operation. (*b*) What kind of coal is very suitable to use with this stoker? Why?

50. (*a*) What are the two outstanding characteristics of the chain-grate stoker? (*b*) What are its disadvantages?

51. Discuss the significance of the terms *chain grates* and *traveling grates.*

52. (*a*) Explain why the leakage of air may be serious with traveling grate stokers. (*b*) Explain how this leakage is minimized.

53. (*a*) How does the thickness of the fuel bed vary from front to rear of a chain-grate stoker? (*b*) Discuss the effect of this variation, and how complete combustion may be attained without excessive amounts of air. (*c*) Explain how to regulate the thickness of the fuel bed at the front.

54. (*a*) Explain what is meant by "tempering" a coal. (*b*) What are the advantages and disadvantages of "tempering"?

55. (*a*) Chain-grate stokers with natural draft are best suited to burn what kinds of coal? (*b*) The design of the arch requires consideration of what factors? (*c*) How many pounds of coal per hr. per sq. ft. of grate may generally be most efficiently handled with this type of stoker? (*d*) What would be reasonable values of the draft, in inches of water, required for the combustion rates in (*c*)?

56. Discuss the forced-draft traveling grates as to (*a*) the kind of fuel that may be used and (*b*) the method of controlling the air to each portion of the grate.

57. (*a*) Explain how some caking coals may be successfully burned on a chain-grate stoker. (*b*) Discuss the curves given by (*a*), (*b*), and (*d*) of Fig. 571.

58. What are the main advantages of front and rear arches, as shown in Fig. 574?

59. Explain the purpose of water-cooled arches and side walls.

60. Discuss the test results shown by the curves in Figs. 576 and 577.

61. What type of traveling grate may be used for small-size anthracite? Why?

62. Discuss the arrangement of arches and settings shown in Figs. 578 to 582 inclusive.

63. (*a*) Explain how to operate a forced-draft traveling grate to obtain the best results with low-volatile fuels. (*b*) The average and maximum combustion rates, in lb. per hr. per sq. ft. of grate, are about how much with anthracite and with coke breeze?

64. (*a*) What kinds of fuel may be burned with underfeed stokers? (*b*) Describe fully the operation of this type of stoker. (*c*) Are arches necessary with this type? Why? (*d*) Are high rates of combustion possible? Why?

65. Define the following terms used with underfeed stokers: (*a*) hoppers, (*b*) rams, (*c*) retorts, (*d*) tuyeres, (*e*) dead plates, and (*f*) dump plates.

66. Make a sketch to show the essentials of (*a*) a simple horizontal single-retort stoker, (*b*) a single-retort stoker with lateral grates, and (*c*) an inclined multiple-retort underfeed stoker.

67. For each class of underfeed stoker in problem 66, discuss the following: (*a*) the method of regulating the fuel and air used, (*b*) the movement of the coal on the stoker, (*c*) the method of removing the refuse, (*d*) the fuel-burning capacity, and (*e*) the usual applications.

68. Answer the following questions concerning the multiple-retort underfeed stoker: (*a*) What are some of the advantages of this stoker compared with other types, when used in large units? (*b*) How may the desired thickness and contour of the fuel bed be maintained as desired? (*c*) Why are overfeed grates sometimes used? (*d*) How is the crank-shaft speed regulated? (*e*) What is the purpose of a shearing-pin? (*f*) What kind of ash-discharging devices are used?

69. (*a*) Make a sketch of a double clinker grinder. (*b*) Compare the amount of carbon in the refuse obtained with different depths of grinder pockets and different rates of operation; compare with dump grates.

70. (*a*) When underfeed stokers are to be operated at high rates, what are the two important factors to be considered in selecting the materials and type of construction to be used in the furnace walls? (*b*) Describe some of the walls that are used.

71. In the operation of multiple-retort underfeed stokers, discuss the (*a*) rates of combustion, (*b*) amount of excess air required, (*c*) windbox pressure, (*d*) effect of preheated air, (*e*) production of fine cinder, and (*f*) power required to operate the stoker.

72. Discuss the performance curves given in Fig. 591.

73. What are the items that should be considered in selecting the proper fuel-burning equipment in any specific case?

74. What type of stoker is generally most suitable for burning (*a*) anthracite, (*b*) semi-bituminous, and (*c*) free-burning bituminous coal containing 6 per cent ash that fuses at 2200 F?

75. What factors, in addition to the fuel, should be considered in determining the proper size and kind of stoker to use?

CHAPTER XXXII

1. (*a*) What are the principal advantages resulting from the use of pulverized fuel? (*b*) The chief disadvantages?

2. Using Fig. 593, trace the fuel (*a*) from the bunkers to the pulverized fuel bin and (*b*) from this bin to the furnace.

3. Explain how a fine particle of pulverized coal is probably ignited and burned in a furnace.

4. Explain how to obtain high combustion rates and short lengths of flame with powdered coal.

5. When burning pulverized coal, what items are affected by the size of particle?

6. What factors affect the *ignition lag* involved in burning powdered fuel?

7. (*a*) Distinguish between *turbulence* and *mixing* as applied to the air and fuel particles in a furnace. (*b*) Which one is of the greater importance? Why?

8. Discuss the relation between furnace volume and completeness of combustion of powdered coal.

9. Explain why the velocity of flame propagation in a mixture of air and powdered coal must be considered in burner design.

10. How is the velocity of flame propagation affected by (*a*) fineness, (*b*) composition of the dry fuel, (*c*) moisture, and (*d*) ratio of primary air to fuel?

11. (*a*) Name seven factors affecting the length of flame, when burning powdered coal. (*b*) Does the length of flame determine the distance from the burner outlet to the tip of the invisible flame? Why?

12. (*a*) What fuels can be burned in powdered form? (*b*) Can the same combustion equipment be used for the various fuels? Why?

13. Discuss the important points involved in the combustion of powdered coal if the fuel is (*a*) anthracite, (*b*) coke, (*c*) coking coal, (*d*) free-burning coal, (*e*) Western sub-bituminous, and (*f*) lignite.

14. Discuss the effect of different coals on the capacity, wear, and energy consumption, of the pulverizing mill.

15. What factors are involved in determining the proper degree of pulverization to give the best economic results?

16. Explain how to express the degree of pulverization of a sample of powdered coal.

17. What degree of pulverization, approximately, will give the best economic results when using (*a*) anthracite, (*b*) Eastern high volatile, and (*c*) sub-bituminous coal?

18. (*a*) What equipment is included in the coal preparation plant using the "storage system"? (*b*) What other names are used to designate this system? (*c*) What are the chief advantages and disadvantages of this system? (*d*) Trace the coal through the system shown in Figs. 597 and 598.

19. (*a*) Describe the "unit system." (*b*) What are its advantages and disadvantages relative to the storage system? (*c*) Trace the coal through the system shown in Fig. 545 (p. 453).

20. Under what conditions may a "combination system" be the best one?

21. Give the desirable characteristics of a good pulverized-fuel burner.

22. What factors affect the design and operation of a burner?

23. What part of the flame is likely to be furthest from the burner? Why?

24. Should a furnace be equipped with one large burner or a number of smaller ones having the same total capacity? Why?

25. Describe the operation of a "stream-line burner."

26. (*a*) What is a turbulent burner? (*b*) Describe the circular burner. (*c*) Describe the intertube burner.

27. (*a*) What is the primary purpose of a "feeder"? (*b*) Is it required with both systems of using powdered coal? Why?

28. Is there any relation between the burner and furnace arrangement? Why?

29. (*a*) Explain what is meant by "vertical firing." (*b*) Where is the secondary air introduced? (*c*) What advantages are gained by providing a small horizontal burner for each vertical burner?

30. Describe "horizontal firing."

31. (*a*) What is meant by "tangential firing"? (*b*) What are its advantages? (*c*) What kind of furnace walls must be used? Why?

32. What are the approximate values of the maximum allowable rates of heat liberation in furnaces of various types of construction?

33. What types of furnace construction may be used with powdered coal? Why?

34. What are the disadvantages of having considerable surface moisture in coal that is to be pulverized?

35. (*a*) Describe some of the ways of drying coal before pulverizing, and also while being pulverized. (*b*) Which system of drying is now preferred? Why?

36. (*a*) Trace the paths of the hot gases and coal through the apparatus shown in Fig. 611. (*b*) What factors affect the amount of drying required?

37. (*a*) What are the desirable qualities of a good pulverizer? (*b*) How may the degree of pulverization be controlled? (*c*) Through what apparatus should the coal pass before delivery to pulverizing elements of the mill? Why?

38. What are the common means employed in mills to pulverize the coal?

39. Describe the operation of one of the mills shown in Figs. 615 to 619.

40. What are the most important factors affecting (*a*) the size of the mill required, (*b*) the power required to drive it, and (*c*) its wear?

41. (*a*) What determines the degree of fineness to which a coal should be ground? (*b*) What is Rettinger's Law?

42. Explain why there is such a wide range of values shown in Fig. 620.

43. What are the difficulties encountered in the storage of powdered coal?

44. Describe the principal systems used to transport powdered coal.

45. (*a*) What factors affect the hazards involved in using powdered coal? (*b*) What precautions should be taken to prevent explosions in the coal-preparation room?

46. (*a*) In a powdered-coal furnace what portion of the ash is deposited in the "furnace bottom"? (*b*) How is this deposit removed from the different forms of "furnace bottoms"?

47. (*a*) What are the troubles caused by fly ash from powdered coal? (*b*) How are these troubles minimized?

48. Discuss the results shown in Fig. 623.

49. (*a*) What are the largest items of cost in preparing pulverized coal? (*b*) What is a reasonable estimate of the total preparation cost?

50. Summarize the main points to be considered in deciding whether powdered-coal or stoker-fired units should be selected.

CHAPTER XXXIII

1. (*a*) To what extent, approximately, is gas now used in the central power stations in this country? (*b*) What kind of gas is commonly used for this purpose? Why?

2. (*a*) What is the ignition temperature of a gas? (*b*) What are the factors upon which this temperature depends? (*c*) Give the approximate value of the ignition temperature of some gas in air at atmospheric pressure, and explain how such a result may be experimentally determined.

3. (*a*) What is meant by the term, *normal velocity of flame propagation*? (*b*) What factors affect this velocity? (*c*) Show by a diagram how this velocity varies with the richness of the mixture.

4. (*a*) Find the proportion, by volume, of natural gas in a mixture of this gas and air to give the maximum flame speed, if the gas analysis shows the following percentages by volume: hydrogen 3, methane 65, ethane 30, nitrogen 1, and carbon dioxide 1. (*b*) Calculate the maximum speed of uniform flame movement for the mixture of gas and air in (*a*). Ans. 8.7 per cent. (*b*) 2.54.

5. The same as problem 4, except the natural gas contains 68 per cent methane and no hydrogen. Ans. (*a*) 8.57 per cent. (*b*) 2.45 per cent.

6. (*a*) Explain by the aid of a sketch how an orifice in a pipe line through which a combustible gaseous mixture is flowing may prevent the passing of a flame through the entire length of the line. (*b*) Under what conditions would such a restriction not act as a barrier to the progress of the flame? Why? (*c*) Discuss the application of (*a*) and (*b*) to a gas burner.

7. If a large portion of the combustion air used in a gas-fired furnace is supplied through the furnace walls or bottom where should the tuyere-blocks be placed? Why?

8. Explain how the degree of mixing of the gas and air within a burner affects the resultant flame.

9. Explain how to obtain high flame temperatures with a gaseous fuel.

10. Describe a gas burner of the (*a*) aspirating type and (*b*) venturi type.

11. (*a*) In gas-fired furnaces what is the approximate range in the rate of energy liberation? (*b*) What are some important effects of having non-luminous flames in the furnace?

12. (*a*) Trace the air and fuels through the unit shown in Fig. 629. (*b*) Why are furnaces sometimes equipped to burn more than one kind of fuel?

13. (*a*) Assuming a fuel to be a mechanical mixture of carbon and hydrogen, derive the equation for its gross heating value. (*b*) Does this equation hold strictly for real fuels? Why?

14. Derive the equation for the weight of the dry products of combustion of a pound of hydrocarbon fuel when burned completely with a known percentage of excess air.

15. A fuel mixture consists of 20 per cent hydrogen and 80 per cent carbon, by weight. This fuel and the combustion air are supplied to a steam-generating unit at a temperature of 90 F. (*a*) If the exit gas temperature is 400 F, find the maximum possible efficiency of this unit when the excess air is 10, 20, 30, 40, and 50 per cent. (*b*) The same as (*a*), except the exit-gas temperature is 500 F. (*c*) The same as (*a*), except the exit-gas temperature is 600 F. (*d*) Plot the results of (*a*), (*b*), and (*c*) with the excess air as abscissa. (*e*) Explain why these curves represent the maximum possible efficiencies for the given conditions.

Ans. (*a*) 85.6, 85.1, 84.6, 84.1, 83.6 per cent. (*b*) 83.6, 82.9, 82.2, 81.5, 80.8 per cent. (*c*) 81.4, 80.6, 79.8, 78.9, 78.0 per cent.

16. (*a*) If the method of calculating the heating value of the fuel in the previous problem gave too high a value, how would such an error affect the efficiencies? (*b*) If the temperature of the fuel and air had been 70 F instead of 90 F, how much, approximately, would the efficiency curves have been affected?

17. (*a*) What is the significance of the curves shown in Fig. 633? (*b*) How does the efficiency curve in this figure compare with the test results obtained in one case where both oil and natural gas were used? (*c*) Is the assumption of the same percentage of excess air entirely fair when attempting to estimate the probable results expected from the actual use of coal, oil, and gas? Why?

18. (*a*) Compare the luminosity from a natural-gas flame with that from oil. (*b*) This difference has what effect on the steam-generating unit?

19. (*a*) What is meant by *surface combustion*? (*b*) What are its advantages?

20. (*a*) Explain how *submerged combustion* of a gas may be obtained. (*b*) What are some applications of this method of combustion?

CHAPTER XXXIV

1. What are the principal applications of oil-burning furnaces?

2. (*a*) What are the chief advantages that result from burning oil instead of coal in a furnace? (*b*) The disadvantages?

3. What are the four problems that must be solved in using oil as a fuel?

4. Describe the essential features of a good storage tank installed for oil fuel.

5. Where are strainers placed? Why?

6. (*a*) What determines the delivery pressure of the oil pumps? (*b*) What types of oil pumps are generally used? (*c*) Is a steady pressure at the burner desirable? Why? How is it obtained?

7. (*a*) Describe a method of preheating fuel oil. (*b*) Why is preheating necessary? (*c*) Why is a by-pass around the heater generally installed? (*d*) What causes the chief resistance to heat transfer between condensing steam and oil?

8. Give briefly the main theories regarding the ignition of oil.

9. (*a*) What are the main requirements of an oil burner? (*b*) Why are *atomizing* *burners* so much more extensively used than *vaporizing* ones? (*c*) How is the atomization effected?

10. (*a*) Explain, by the aid of sketches, what is meant by *outside-mixing* and *inside-mixing burners*. (*b*) Describe the kind of flame produced by each of these burners. (*c*) What are the advantages and disadvantages of *steam-atomizing burners*? (*d*) What are the chief applications of the steam-atomizing burners?

11. What are *mechanical-atomizing burners*?

12. (*a*) Describe the *spray-nozzle burner*. (*b*) What are its chief applications? Why? (*c*) Discuss the methods of regulation used with it.

13. (*a*) How have many hand-fired coal furnaces been fitted with steam-jet oil burners at small expense? (*b*) What have been the main difficulties experienced? (*c*) How have these difficulties been overcome?

14. (*a*) Compare the spray from a mechanical-atomizing burner with that from the steam-jet type. (*b*) How may very high rates of energy liberation per unit of furnace volume be secured?

15. Describe the methods of admitting air to the oil-fired furnace using (*a*) flat-spray steam-atomizing burners, (*b*) steam-jet burners that give a conical flame, and (*c*) mechanical burners.

16. (*a*) What has experience shown regarding refractory walls in oil-fired furnaces? (*b*) What have careful measurements shown regarding furnace temperature and slagging action in such furnaces?

CHAPTER XXXV

1. What are the main advantages and disadvantages of the carburetor type of internal combustion engine?

2. (*a*) Why is so much effort expended to develop the high-speed engine using auto-ignition of the fuel sprayed into the cylinder by mechanical means alone? (*b*) Discuss the suitability of the various names used to designate such an engine.

3. What are the objections to air-injection of the fuel in automotive engines?

4. (*a*) Define *ignition temperature*, as applied to the fuel used in an oil engine. (*b*) Upon what does this temperature depend? (*c*) Compare the ignition temperature of an oil with that of its vapor.

5. Discuss the fuel factors, except ignition, that are of importance when dealing with airless injection systems.

6. (*a*) Explain what is meant by the *ignition lag* in an oil engine. (*b*) What factors affect this lag? (*c*) What is the order of magnitude of this lag?

7. Give the theories regarding the ignition of oil sprayed into the combustion chamber of an engine.

8. Discuss *detonation* in the high-speed oil engine.

9. (*a*) What are the three important functions of the fuel-injection system? (*b*) What equipment makes up this system? (*c*) Why must some of this equipment be made with unusual precision?

10. (*a*) Calculate the weight of oil delivered to each cylinder per cycle in an eight-cylinder, four-stroke-cycle engine delivering 200 b.h.p., when running at 1500 r.p.m. and using 0.4 lb. of fuel per b.h.p. hr. (*b*) Find the length of the injection period if it occupies 30 deg. of the crank movement.

Ans. (*a*) 0.000222 lb. (*b*) $\frac{1}{300}$ sec.

11. Name and describe the two systems of airless injection of fuels.

12. (*a*) What is the purpose of an injection nozzle? (*b*) What other names are used for it? (*c*) Describe an open nozzle and give its chief advantages. (*d*) What is a closed nozzle, and what are its chief advantages?

13. (*a*) Explain what is meant by *injection lag*. (*b*) Discuss Rothrock's results, as given in Fig. 644.

14. (*a*) What are the three main factors that determine the weight of oil discharged from an injection nozzle? (*b*) Discuss the results shown in Fig. 645.

15. Discuss the results shown in Fig. 646.

16. (*a*) Using the data as given in problem 10, calculate the diameter of the orifice in the injection nozzle if the coefficient of discharge is 0.66, the density of the fuel 53 lb. per cu. ft., the compression pressure 500 lb. per sq. in., and the injection pressure 3500 lb. per sq. in. (*b*) The same as (*a*), except the injection pressure is 4500 lb. per sq. in. (*c*) The same as (*a*), except the coefficient of discharge is 44 per cent greater. (*d*) Make a sketch to show the shape of an orifice that would make possible such a discharge coefficient as that in (*c*).

Ans. (*a*) 0.022 in. (*b*) 0.0205 in. (*c*) 0.0184 in.

17. For a given injection nozzle, fuel, combustion chamber, chamber pressure and temperature, would an increase in the injection pressure cause an increase, or decrease, in (*a*) the fineness of atomization, (*b*) the penetration in 0.001 sec., (*c*) the ignition lag, (*d*) the rate of combustion, and (*e*) the possible speed of the engine without reducing the completeness of combustion?

18. For an engine having injection during a constant angular movement of the crank shaft, discuss the effect of changing the injection pressure on (*a*) the power delivered by the engine, (*b*) the thermal economy of the engine, and (*c*) the first cost and maintenance of the injection system.

19. (*a*) Name and describe the three main types of combustion chambers used with high-speed oil engines. (*b*) What are the main advantages and disadvantages of each type?

20. Discuss the important features of the combustion chambers and injection nozzles shown in (*a*) Fig. 648, (*b*) Fig. 649, and (*c*) Fig. 650.

21. Discuss the performance curves shown in Fig. 651.

22. From the curves given in Fig. 652, for half load and full load, determine (*a*) the mechanical efficiency of the engine and (*b*) the ratio of the maximum cylinder pressure to the indicated m.e.p.

23. Give the reasons for the change in the appearance of the exhaust products, as indicated in Fig. 652.

24. Discuss the use of *pressure-volume*, and *pressure-time*, diagrams in studying high-speed oil engines.

25. What part of the data, as given in Table LVII, do you consider to be the most significant? Why?

CHAPTER XXXVI

1. (*a*) Describe several ways in which the by-product gas from a blast furnace may be utilized. (*b*) Explain why the combustion space necessary for a given capacity must be large when blast-furnace gas is burned in steam-generating units.

2. (*a*) Describe the arrangement of a typical furnace for burning wet wood. (*b*) In the adiabatic combustion of dry wood with no excess air, what temperature might the products of combustion attain? How does this temperature influence furnace design?

3. What two advantages are derived from the mechanical atomization of refinery waste sludge as compared with steam atomization?

4. (*a*) What is meant by the term *mixed refuse*? Why is this mixing usually necessary? (*b*) Describe the uses of the heat evolved from the combustion of city waste. (*c*) Describe the construction of the furnace of a typical incinerator. (*d*) How is sewage gas produced, and what is its chief constituent?

5. (*a*) What are the approximate heating values of sewage gas, wet garbage, and dry garbage? (*b*) Give some examples to illustrate how the energy obtainable from the burning of city refuse may be utilized to advantage.

CHAPTER XXXVII

1. (*a*) What are the three purposes of controlling combustion in a steam-generating unit? (*b*) Explain why, in general, the proper proportion of air and fuel is difficult to maintain.

2. (*a*) Explain why the furnace draft is so important in any combustion control system. (*b*) What is meant by a "balanced draft"?

3. (*a*) Assuming that the steam gages used are without error, will the one on the steam drum read the same as the one at the outlet of a steam-generating unit? Why? (*b*) Explain why the variation of the outlet pressure may be used to regulate the rate of firing.

4. Explain how quick adjustments of the combustion rate may be obtained with various fuels.

5. (*a*) What is the purpose of the Bailey Boiler Meter? (*b*) Distinguish between *hand control*, *remote control*, and *automatic control*. (*c*) What are the main advantages and disadvantages of remote control as compared with hand control?

6. (*a*) What are the main advantages of an automatic combustion control system? (*b*) What are the primary actuating forces that may be utilized in such a system? (*c*) Describe the main features of some automatic control system.

7. (*a*) What is a *banked fire*? (*b*) A dead bank? (*c*) A floating bank?

8. (*a*) What is meant by the *banking loss*? (*b*) This loss is least with what fuels? Why? (*c*) What is roughly the amount of coal required per hour per square foot of grate for a banking period of 10 hr., and also for 40 hr.? (*d*) In bringing a unit back to normal operating conditions after banking, what systems of firing require the least time? Why? What is the approximate magnitude of this time with large units banked overnight?

CHAPTER XXXVIII

1. (*a*) What are the *primary purposes* of the furnace in a steam-generating unit? (*b*) Is the design of such a furnace important? Why? (*c*) What are the main considerations affecting the design of such a furnace? (*d*) What are the advantages of a high rate of energy liberation per unit of furnace volume?

2. (*a*) What are the four usual kinds of furnace walls? (*b*) What limits the maximum temperature of the reacting agents in a furnace that is to be operated without deterioration? Discuss. (*c*) Why are nearly all large furnaces specially designed for the unit of which they are a part?

3. (*a*) What is meant by the term "specific energy release rate"? (*b*) What factors determine a safe value of this rate? (*c*) Give some conservative values of the specific energy release rates that should not be exceeded with the different types of furnace walls, when burning powdered coal. (*d*) Would the values given in (*c*) be increased with furnaces fired by stokers, or by hand? Why?

4. Discuss fully the results shown by the curves in Fig. 656.

5. For the same rate of energy liberation, should the furnace for burning blast-furnace gas be of the same size as that for natural gas? Why?

6. Are all parts of the furnace walls at approximately the same temperature? Why?

7. (*a*) Describe the main features of simple furnace walls suitable for H.R.T. boilers. (*b*) What is the purpose of the relieving arch? (*c*) Describe the monolithic lining and wall of a furnace.

8. Describe the furnace walls illustrated in Fig. 660.

9. Describe the main features of the walls shown in Fig. 662.

10. (*a*) What are the parts of a furnace wall that require special consideration in stoker-fired furnaces? Why? (*b*) Explain how to obtain satisfactory joints in refractory walls.

11. (*a*) Describe the air-cooled walls shown in Figs. 664 and 665. (*b*) Why are air-cooled blocks often used in furnace walls? (*c*) What limits the height of a self-supporting furnace wall? (*d*) How may tall, hollow, firebrick walls be made?

12. (*a*) What are the two types of arches used in furnace construction? (*b*) Which type is now generally preferred? Why?

13. What are the inherent limitations of refractory walls?

14. (*a*) Under what conditions should water-cooled furnace walls be used? Why? (*b*) Why are some furnaces made with walls partly of refractories and partially water-cooled? Give typical examples.

15. (*a*) Name the three classes of water walls. (*b*) Explain briefly the chief features of each class. (*c*) Describe two methods used to obtain nearly continuous water walls with bare tubes.

16. Explain how, in the covered-tube type of wall, the blocks are fastened to the tubes shown in Figs. 678 to 682 inclusive.

17. Discuss the effect of the flow of molten ash over the refractory facing of water-cooled walls.

18. (*a*) Describe some of the refractory-protected water tube walls. (*b*) What are the main advantages of these refractory coverings?

19. (*a*) The arrangements and types of furnace bottoms depend chiefly upon what factors? (*b*) Describe the types used with stokers; with oil; with gas.

20. (*a*) Explain why furnace bottoms used with powdered coal require special care in their design. (*b*) Name the two types. (*c*) Describe each type. (*d*) What are some of the advantages of the slagging type of furnace?

21. (*a*) What are the serious disadvantages of sluggish circulation in water walls? (*b*) What factors tend to increase the circulation? (*c*) How may *equal distribution* to all tubes be obtained? (*d*) Describe the circulation through the water walls shown in Figs. 688, 689, and 691. (*e*) How are localized stresses in these walls avoided?

22. (*a*) Name seven desirable properties that should be possessed by refractories. (*b*) Which two of these are of the most importance?

23. (*a*) Discuss fireclay, as a refractory. (*b*) For what is silicon carbide often used? Why?

24. (*a*) Name the five ways in which a refractory may fail. (*b*) Discuss each one.

25. (*a*) Name a refractory whose thermal conductivity increases with its temperature; also one for which the opposite is true. (*b*) Is very high thermal resistance always of the utmost importance in selecting a refractory? Why?

26. What are the main points indicated by the Phelps Chart given on page 642?

27. A layer of insulation applied to the outer surface of a refractory furnace lining has what effect on (*a*) the temperature of the refractory, and (*b*) the rate of heat transmitted through this refractory.

28. (*a*) Name the six desirable characteristics of high-temperature insulators. (*b*) What are the four basic materials used for such insulators? (*c*) Why are these materials good insulators?

29. (*a*) What are refractory insulators? (*b*) What material is largely used for this purpose?

30. Discuss the energy streams, as given in Fig. 696.

31. (*a*) Define the term, *the fraction cold.* (*b*) Find the value of this fraction for a cubical furnace, stoker fired, with roof and three walls water-cooled.

32. If a powdered fuel furnace, 20 by 20 by 20 ft., should have energy liberated within it at the rate of 70,000 B.t.u. per hr. per cu. ft. of flame volume, what would be the probable rate of heat absorption by the furnace walls (*a*) if $\psi = 1$, and (*b*) if $\psi = \frac{1}{3}$.

Ans. (*a*) 58,000 B.t.u./hr./sq. ft. (*b*) 150,000 B.t.u./hr./sq. ft.

33. The same as problem 32, except the rate of energy liberation within the furnace is 10,000 B.t.u. per hr. per cu. ft. of flame volume.

Ans. (*a*) 15,700 B.t.u./hr./sq. ft. (*b*) 39,500 B.t.u./hr./sq. ft.

34. Discuss the test results shown by the curves in Fig. 698.

35. What is shown by the curve in Fig. 699?

36. What is the significance of curves similar to that in Fig. 700?

37. Discuss the effect of the different types of construction of solid walls, as shown in Fig. 701, on (*a*) the rate of heat transmitted through the wall, and (*b*) the temperature gradient through the refractory.

38. Calculate the amount of heat transmitted through a furnace wall having a surface of 1200 sq. ft. and a hot-surface temperature of 2000 deg. fahr. The wall is made of 9 in. of firebrick and 4.5 in. of super Sil-O-Cel insulation. Assume the room temperature to be 80 deg. fahr. Ans. 624,000 B.t.u./hr.

39. Illustrate the use of the curves in Figs. 702 and 703.

40. Explain why *dead air spaces* are not used in modern furnace walls.

41. What are the factors that determine the difference between the temperature of the exposed surface of a furnace wall and that of the water circulating within it.

42. Discuss the results shown by the curves in Fig. 704.

43. (*a*) In an air-cooled refractory wall, what is the chief factor controlling the rate of heat transfer from the refractory to the air? (*b*) What determines the equilibrium temperature of the cool surface of the refractory at various sections of the air passage?

44. Discuss the results of Kolb, as shown in Figs. 705 and 706.

45. (*a*) Give the assumptions on which the results of Figs. 707 to 711 are based. (*b*) Give the meaning of each symbol used. (*c*) Illustrate the use of the curves in Fig. 707; in Fig. 708; in Fig. 709; and in Fig. 710.

46. When burning pulverized coal, what is the effect on the flame temperature of increasing the furnace volume 50 per cent, but keeping the same air temperature and the same rate of energy release?

47. The same as problem 46, except that the furnace is stoker fired.

48. What influence does the type of coal have on the flame temperature, and on the rate of heat transfer from a powdered coal furnace?

49. Discuss the effect of adding a refractory coating to the water-wall surface of a furnace.

50. What is the *fundamental principle in furnace design*, as given by Mr. Bailey?

CHAPTER XXXIX

1. (*a*) What was the meaning of the term *boiler* as often used formerly? (*b*) What is its present meaning?

2. What elements constitute the usual boiler structure?

3. (*a*) How is the *general suitability* of a boiler determined? (*b*) How may its *safety* and *durability* be estimated? (*c*) Why is *accessibility* necessary?

4. In selecting a boiler, what points should be noted regarding (*a*) water circulation, (*b*) gas passages, and (*c*) priming?

5. (*a*) Where should the feedwater enter a boiler? (*b*) What are some important points that affect the space required for a steam-generating unit? (*c*) What items must be considered in estimating the cost of a steam-generating unit?

6. (*a*) On what bases may boilers be classified? (*b*) Why is the pressure an unsatisfactory basis?

7. (*a*) What are the three prime advantages of the H.R.T. boiler? (*b*) What sizes and pressures are ordinarily used? (*c*) Describe the main structural features of this boiler.

8. (*a*) Describe the small internally fired tubular boiler of the submerged-tube type, and of the exposed-tube type. (*b*) What are the chief advantages and disadvantages of this boiler?

9. What sizes and pressures are generally used in "large vertical tubular boilers"?

10. Describe the locomotive boiler and furnace.

11. (*a*) How are modern coal-burning locomotives generally fired? (*b*) What are the average combustion rates likely to be? (*c*) Are these high rates feasible? Discuss fully all factors involved.

12. (*a*) What are the advantages of the arch tubes shown in the furnace of Fig. 720? (*b*) What are the advantages gained by placing the refractory on these tubes?

13. What are the distinctive features of each of the following: (*a*) Stationary locomotive boiler? (*b*) Scotch marine boiler? (*c*) Dry-back Scotch boiler?

14. (*a*) Describe the B. and W. longitudinal-drum boiler. (*b*) What are the merits of the vertical headers as compared with the inclined ones? (*c*) Explain how the hand holes are closed. (*d*) What sizes and pressures are ordinarily found in this type of boiler?

15. (*a*) Describe the box-header type of longitudinal-drum boiler, or Heine type. (*b*) Describe the methods of staying the throat and the box headers.

16. (*a*) Describe a cross-drum boiler. (*b*) Compare its advantages with the longitudinal type.

17. Discuss the main features of the boilers shown in (*a*) Fig. 733, (*b*) Fig. 734, and (*c*) Fig. 735.

18. What are the advantages of the single-pass cross-drum boiler?

19. Using the data given in Tables LXV and LXVI, for a steam flow of 273,000 lb. per hr., find (*a*) the mean specific heat of the gases in passing through the air heater, and (*b*) the overall coefficient of heat transmission of this heater.

Ans. (*a*) 0.252. (*b*) 3.79 B.t.u./hr./sq. ft./deg. fahr.

20. What are the principal characteristics of the following boilers: (*a*) Wicks vertical? (*b*) Bigelow–Hornsby semi-vertical? (*c*) Edge Moor single pass?

21. (*a*) What are the main advantages of the bent-tube type of water-tube boilers? (*b*) What are the advantages and disadvantages of using four drums? (*c*) What are the advantages and disadvantages of using large drums? (*d*) How are the drums supported? (*e*) Can these boilers be arranged in continuous batteries? Why?

22. (*a*) Describe the circulation of the water in the Stirling type of boiler. (*b*) Describe the baffling that may be used with this boiler.

23. What are the main features of the boilers represented in Figs. 744 and 745?

24. Describe the *combustion steam generator*.

25. (*a*) Describe a large double set steam-generating unit. (*b*) Using the data given for the design of the Hell-Gate unit, shown in Fig. 749, calculate the tons of coal burned per hour by it when delivering 800,000 lb. of steam per hr., assuming the coal to have a heating value of 12,300 B.t.u. per lb. (*c*) What is the weight of coal burned per hour per cubic foot of furnace for the conditions given in (*b*)?

Ans. (*b*) 45.1 tons. (*c*) 2 lb.

26. Compare the dimensions of the Kips Bay unit with some local building.

27. Explain why *forced circulation* may be desirable in certain types of boilers.

28. (*a*) What are *small tube boilers*? (*b*) What are their advantages and usual applications? (*c*) What is the advantage gained by using the *porcupine tubes* in the mercury boiler?

29. (*a*) What are flash boilers? (*b*) Semi-flash boilers?

30. Describe the unique features of the following boilers: (*a*) Atmos, (*b*) White (automobile), (*c*) Benson, (*d*) Schmidt–Hartman, (*e*) Loeffler, (*f*) Brown–Boveri.

31. (*a*) What are the three types of electric boilers? (*b*) Describe each type. (*c*) Under what conditions are such boilers used?

32. (*a*) What are steam accumulators used for? (*b*) What are the two types? (*c*) Describe each one.

33. (*a*) Under what conditions is the circulation system of a boiler most likely to give trouble? (*b*) Why is good circulation so important? (*c*) Poor circulation may cause what troubles?

34. (*a*) Explain how natural circulation in a boiler is produced. (*b*) When is a boiler said to be *steam bound*?

35. What are the causes of changes in water level in a boiler?

36. (*a*) Describe the manner in which steam is formed in a riser. (*b*) Explain what is meant by *spouting, priming*, and *foaming*.

37. (*a*) What is likely to be the approximate value of the most economic rate of heat absorption per square foot of the total heating surface of a simple boiler? (*b*) Does this same rate generally hold for central station boilers? Why?

38. (*a*) What are the terms commonly used to designate the two classes of heating surfaces of a boiler? Why? (*b*) Under what conditions of operation are these two classes of surfaces likely to absorb about equal amounts of energy?

39. Discuss the temperature gradients shown in Fig. 770.

40. Using the data given in Fig. 770, find the value of the overall coefficient of heat transfer. Assume the area of the heating surface to be 5000 sq. ft., the weight of gases to be 45,000 lb. per hr. and the mean specific heat of the gases to be 0.275.
Ans. 5.31 B.t.u./hr./sq. ft./deg. fahr.

41. What factors are involved in determining the best combination of tube diameter, tube length, and area of gas passages in a boiler?

42. (*a*) What is, approximately, the range of the *average rates* of heat absorption per square foot of heating surface of boilers in the United States? (*b*) The same as (*a*), except that the *maximum rates* are to be used.

43. Explain how to determine the most economical amount of heating surface to install in any given case: (*a*) when only the boiler is involved; (*b*) when boiler and economizer are to be installed; and (*c*) when boiler, economizer, and air heater are to be used.

44. (*a*) Why do the pressure and temperature of the steam at the outlet of a steam-generating unit affect the distribution of the heating surface in such a unit? (*b*) Compare the relative costs of a square foot of heating surface of the different elements in a modern unit.

45. Discuss the temperature curves shown in Fig. 776.

46. (*a*) What are some of the basic requirements of boiler materials? (*b*) What special requirements are to be met when the temperature of the metal is above 700 deg. fahr.?

47. (*a*) How are the *joints* in boiler drums to be made? (*b*) Describe the methods of making the tube holes. (*c*) How are the tube ends fastened to the drums?

48. (*a*) What are boiler accessories? (*b*) Which ones are always required?

49. (*a*) Describe the *pop type* of safety valve. (*b*) What are its advantages? (*c*) What is meant by the *blowdown*?

50. (*a*) How many safety valves are required on large boilers? (*b*) On superheaters? (*c*) How is the total safety valve capacity of a unit calculated? (*d*) What special requirements are involved with safety valves used for *high pressures*, and also for *high temperatures*?

51. (*a*) Discuss fusible plugs, as applied to boilers. (*b*) Describe the water column of a boiler. (*c*) Describe a water gage used for pressures above 1000 lb. per sq. in.

52. (*a*) What is the purpose of a blowoff valve? (*b*) Why is such a valve subject to severe service? (*c*) Describe the main features of a modern blowoff valve. (*d*) Describe the blowoff connections.

53. (*a*) What are the advantages of automatic feedwater regulators? (*b*) What are some of the difficulties encountered with such regulators?

54. Explain the operation of an automatic damper regulator.

55. How are scale and soot removed from heating surfaces?

56. What part of a boiler is most likely to fail? Why?

57. If any part of a steam-generating unit explodes, which of the following items are of the greatest importance in estimating the probable destructive effect: steam pressure, steam temperature, steam space, water space, type of boiler, location of rupture, rate of firing, and kind of fuel used?

CHAPTER XL

1. State the principal uses of separately fired superheaters.

2. By sketches, show typical locations of (*a*) radiant and (*b*) convection intertube types of integral superheaters.

3. Draw and explain the characteristic performance curves of (*a*) radiant, (*b*) convection, and (*c*) combination superheaters.

4. Explain the advantages of the extended-surface type of superheater tube.

5. A convection superheater with a heating surface of 3000 sq. ft. is to operate under the following conditions:

w_s Lb. Steam per Sec.	w_g Lb. Gas per Sec.	θ_m Deg. Fahr.
20	35	300
40	67	520
60	95	680
80	125	750

For these conditions, the film coefficient of heat transfer on the steam side is $h_s = 5w_s^{0.8}$; on the gas side, $h_g = w_g^{0.6}$ B.t.u. per hr. per sq. ft. per deg. fahr.; and the equivalent thickness of the tube with deposits is 0.2 in. with a specific thermal conductivity, $k = 100$. Find, at each load: (*a*) the overall coefficient of heat transfer, (*b*) the temperature rise of the steam in passing through the superheater, if $c_p = 0.6$. Ans. (*a*) 7.21, 10.81, 13.45, 15.81. (*b*) 150.2, 195.1, 212, 206.

6. Explain, for a given fraction cold, the temperature characteristics of a radiant superheater from the curves of Fig. 708 (*a*).

7. (*a*) What is the chief difference in principle between the reheaters shown in Figs. 806 and 807. (*b*) Explain the principal advantages of gas reheating over steam reheating when the steam reheater is located near the turbine.

8. (*a*) Describe the methods used to regulate the reheat temperature. (*b*) Explain the precautions taken to prevent overspeeding of the low-pressure turbine when the steam is reheated.

9. (*a*) In what way may the use of diphenyl simplify the reheating problem? (*b*) Describe the proposed regenerative reheating system and state its advantages over the other reheating systems.

10. (*a*) For what purposes are desuperheaters used? (*b*) How may the steam temperature be regulated in a surface type of desuperheater? (*c*) Describe three types of direct-contact desuperheaters.

CHAPTER XLI

1. Give the *primary* advantages and disadvantages that result from the use of economizers and air heaters.

2. Define *economizer surface* and *air-heater surface.*

3. (*a*) List the important requirements that must be met by an economizer. (*b*) Compare the *group type* with the *unit type* economizer.

4. (*a*) Sketch a typical arrangement of a *group* type economizer and the boilers that it serves. (*b*) Sketch a typical arrangement of an *integral* type economizer. (*c*) Sketch the elements of an *extended surface* economizer.

5. (*a*) Give the *secondary* advantages and disadvantages resulting from the use of air heaters. (*b*) Discuss the factors that limit the temperature to which the air shall be preheated.

6. (*a*) Compare the *recuperative* with the *regenerative* type of air heater. (*b*) Why may it be desirable to have reversible elements in air heaters? (*c*) What is a *supermiser?*

7. (*a*) Explain how the heat rejected from internal-combustion engines may be recovered and used. (*b*) Why, in general, is it more expedient to recover the heat rejected from four-stroke rather than from two-stroke cycle engines?

8. (*a*) What offers the greatest resistance to the flow of heat from gas to water in an economizer? (*b*) How may this resistance be decreased?

9. (*a*) List the resistances to heat transfer from gas to air in an air heater. (*b*) Why, in general, is the overall coefficient of heat transfer slightly lower in an air heater than in an economizer for the same mass velocity of the gas?

10. An economizer operates under the following conditions:

	Load 1	Load 2
Lb. gas per sec. per sq. ft. (w_g/A)	0.833	1.67
Lb. water per sec. (w_w)	7.5	12.5
Mean temp. diff. deg. fahr. (θ_m)	180	200

The film coefficient of heat transfer on the gas side is $h_g = 4.4 \left(\frac{w_g}{A}\right)^{0.6}$; on the water side, $h_w = 20\,(w_w)^{0.8}$; and the equivalent thickness of the tube is 0.25 in. with a specific thermal conductivity, $k = 200$. Find, for each load: (*a*) The overall coefficient of heat transfer. (*b*) The temperature rise of the water in the economizer assuming a specific heat of unity and an economizer surface of 4800 sq. ft.

Ans. (*a*) 3.77, 5.71; (*b*) 120.5, 122.

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3 9358 00018388 6